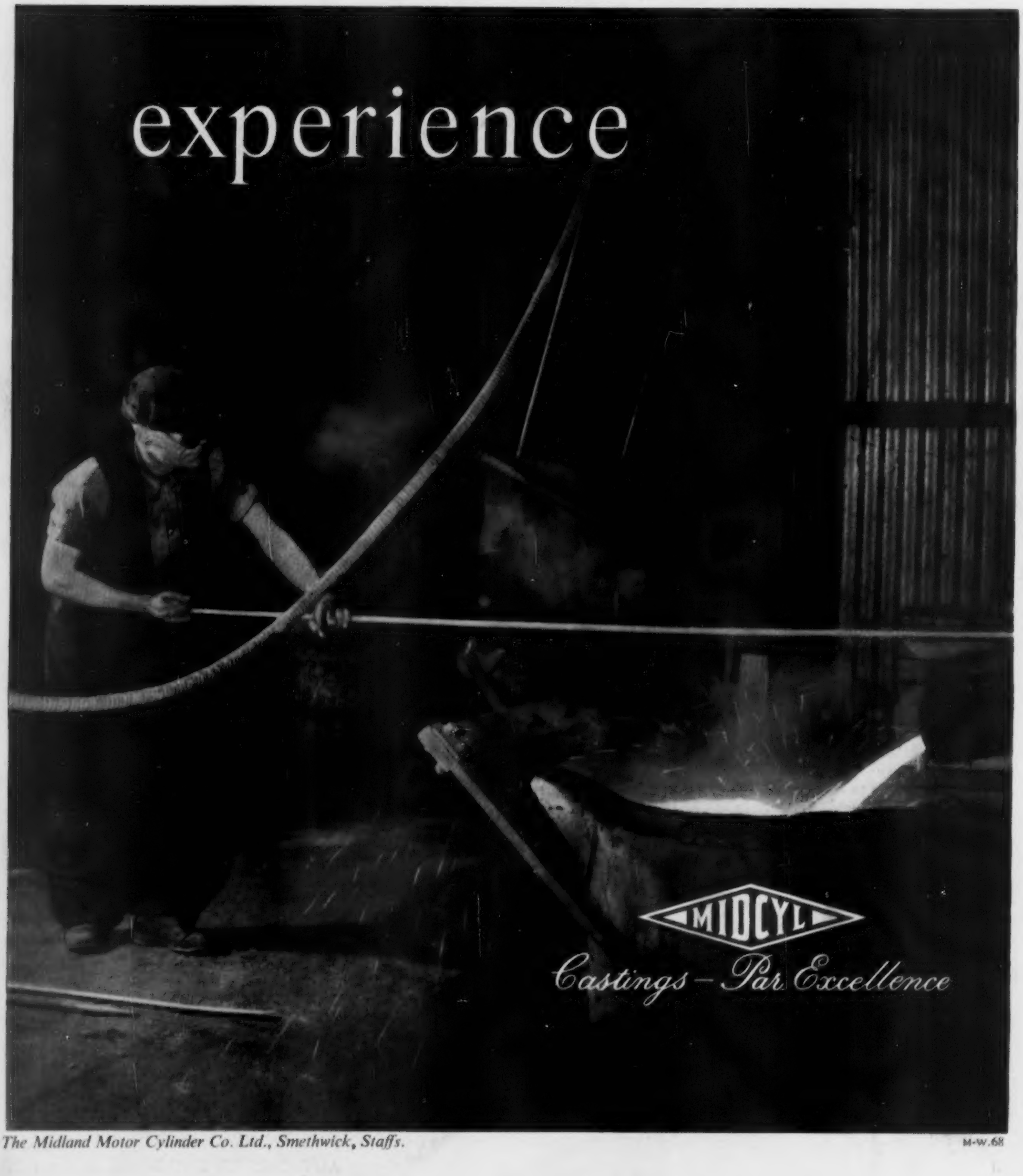
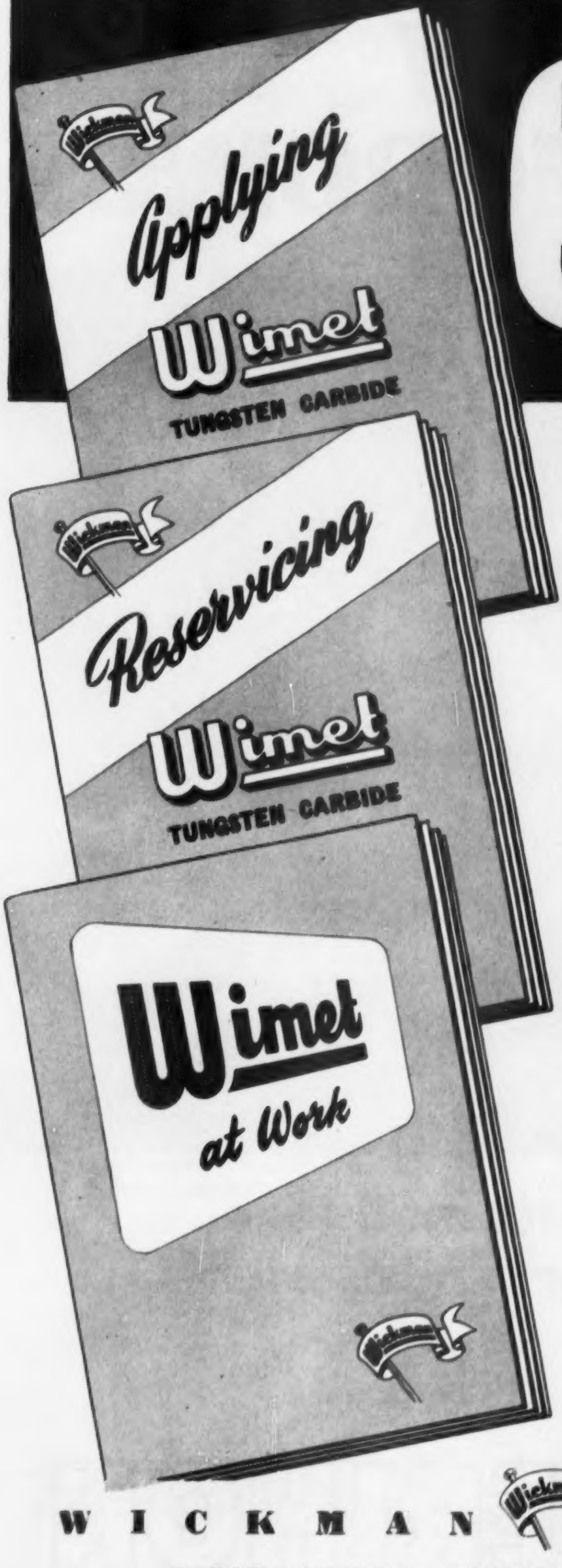
AUTOMOBILE ENGINEER

DESIGN · PRODUCTION · MATERIALS

Vol. 45 No. 8 AUGUST 1955 PRICE: 3s. 6d.





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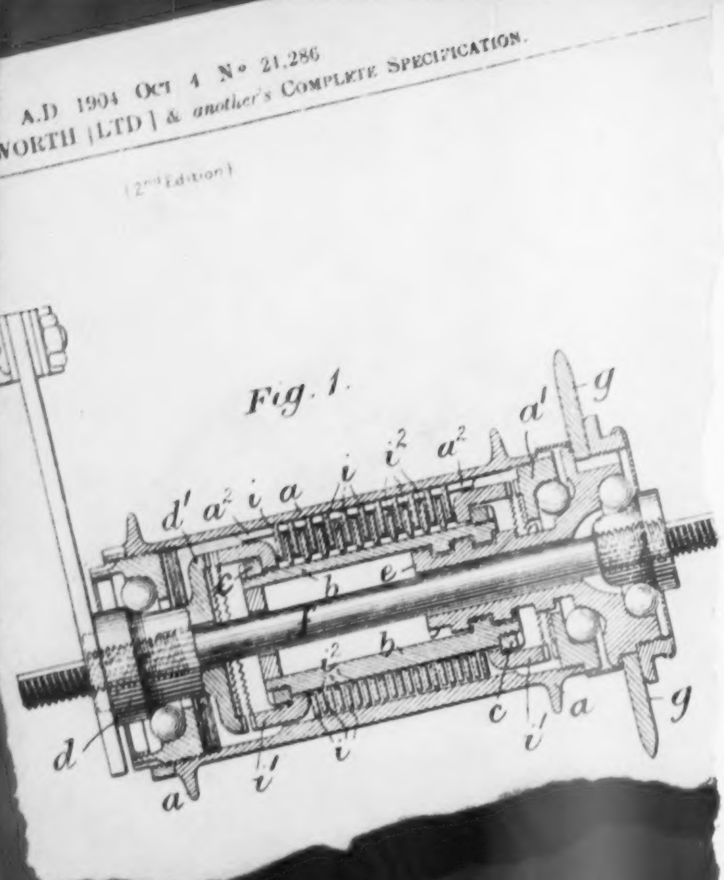


The outer plates are splined to the hub, the inner plates splined to the sleeve; this revolves en bloc with the hub during pedalling or freewheeling. On back-pedalling, the sleeve is arrested by the face clutch teeth: further back-pedalling presses the discs together. One of many neat designs by the late J. V. Pugh of Rudge-Whitworth.

BRAKES: Ways of dissipating the energy

We say 'dissipate' because you can't destroy it but only convert it into other forms of energy, generally heat.

Can we dissipate it electrically? On a drastic scale, no, not without great weight and complication. It is pathetic that it is so difficult to throw away something so valuable. An ingenious test gear, for simulating the effect of mountainous country, was evolved by one famous manufacturer, who towed a trailer whose live axle drove a large dynamo, and by suitable controls and resistances he was able to set up a drag load equivalent to any required gradient. Here resistance grids were arranged to radiate heat to large volumes of air; an ideal procedure for dispersing energy in an accurate test, but of only passing interest to the car owner.

Transmission-driven brakes are in use which utilize the drag set-up when eddy-currents are induced in a member revolving in a magnetic field. These currents in turn generate heat which is dissipated into the atmosphere by an efficient cooling system. Such brakes have the advantage of being free from actual brake shoes or other wearing parts, and easily operated by switching on the excitation current. Their action should be very consistent and well suited to mountain roads. It would seem, however, that at low speeds the braking would have to be supplemented by other brakes, and that the installation would be rather heavy.

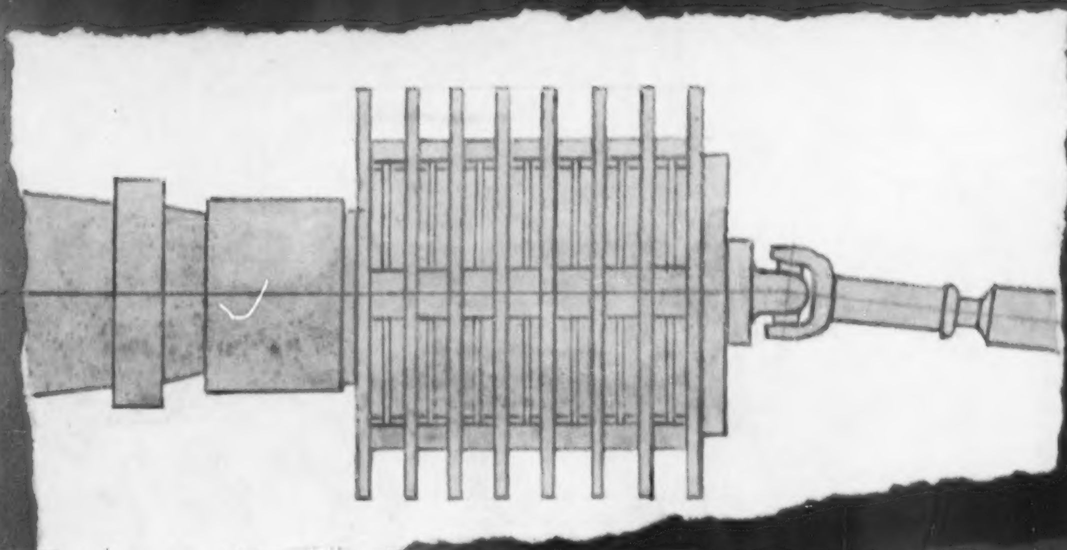
We also see the regenerative braking employed on some electric railways, where the energy is not dissipated, but 'paid back into the bank', which is better still.

In some vehicles the engine is used as a brake—air being compressed by the engine but released by a special camshaft before it can give back its energy to the crankshaft. In other systems, the exhaust is throttled. Here again, the heat is passed to large volumes of air, and in fact, short of utilizing the latent heat of ice or water, as on record-breaking cars, the only convenient way at present for us to discard our unwanted energy is to heat up the air around us, either first-hand, as by an air brake, or through some intermediate step. If weight were no object, we could use our energy to churn up water, as in Joules' calorimeter, or use a friction brake cooled with water, which in either case we could cool in a radiator.

If we wanted to store our energy as mechanical energy, we should have new problems. A new Swiss 'bus has a great fly-wheel, which is wound up electrically, its momentum propels the 'bus to the next stop, where it takes in current from overhead gear and winds it up again. Planned as they have it planned, it works well, but we cannot very well carry an enormous extra fly-wheel on a pleasure car, just in order to be able to wind it up when we want to slow down. Moreover, the amount of energy we should be able to save would be more than expended in carrying the heavy fly-wheel about.

We can't change our spare energy into sound, you get too

SOME BASIC CONSIDERATIONS



much noise for quite a little energy, and other wavelengths are equally unhelpful.

If we decide to stick to mechanical friction, and therefore disperse the energy as heat, then we must have a much larger area to heat a large volume of air. Where can we get it? There isn't much room for larger drums of present type.

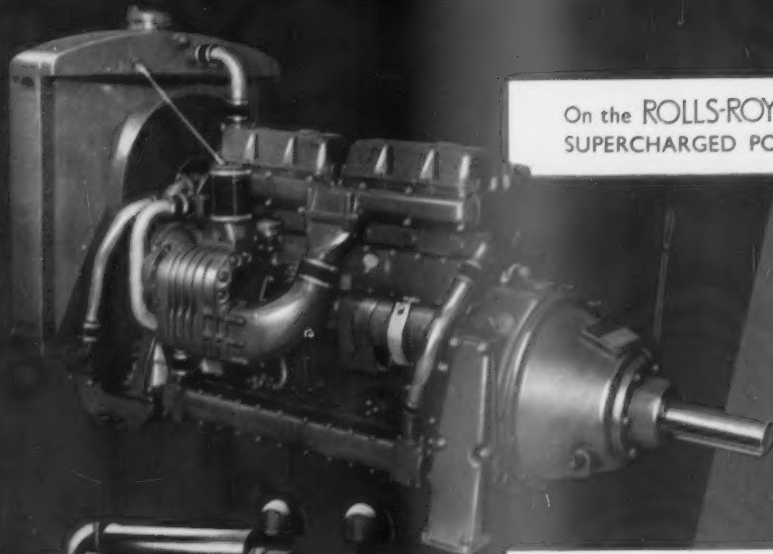
If we accept the idea of using a transmission brake, we find plenty of length available, even if diameter were restricted, because with a car geared 5 to 1 a transmission brake of only 6 inches diameter is equivalent to a brake drum bigger than the road wheel. By 'length' in such a brake we do not mean a conventional brake stretched lengthways along the transmission by using abnormally wide shoes and drum, but a succession of brakes in tandem, either a pack of discs such as Lanchester used in the early part of the century, but out in the open, or a 'battery' of separate disc brakes acting in parallel. It would, of course, be necessary to use a limited slip differential gear, and the transmission would have to be beyond reproach on the overdrive.

It may be objected that such a brake would never be properly 'off', and would always be dragging, but that is not necessarily the case, as is shown by the 50-year-old patent drawing on the left.

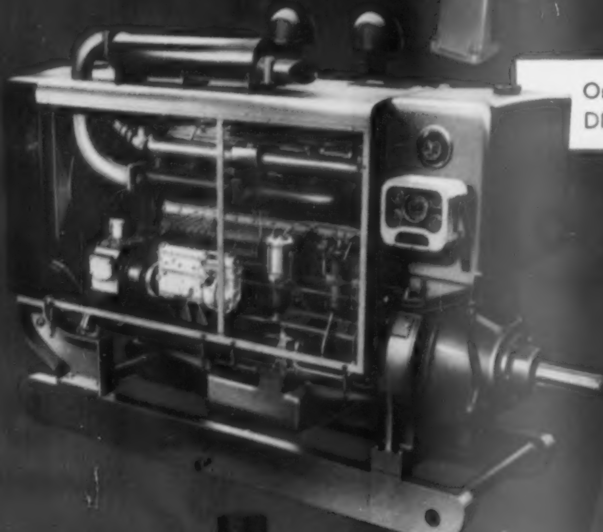
With such mechanisms as synchromesh available to us, plenty of length, and an experimental turn of mind, such a brake can be developed if the occasion warrants it; it is mentioned as suggesting a possible train of thought.

The reader may have realized that, among the alternative forms of energy, we never considered changing our surplus mechanical energy into chemical energy, though we do it on a minute scale if we run downhill in gear, for in charging we set up a chemical change in the battery which enables it to give back some electricity while reverting to its previous chemical state. But at the present state of knowledge the answer is the old one—prohibitive weight and complexity.

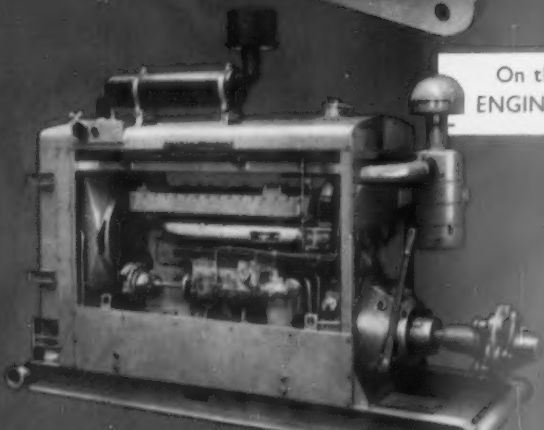
In the meantime, if the reader should discover some very simple and lightweight apparatus which can be fed with a substance which is moderate in cost, absorbs unwanted energy and in doing so perhaps transmutes itself into something more valuable, he will be assured of an enthusiastic welcome in many parts of the world, including Leamington Spa. It would be a kind of negative dynamite, and we might call it 'Dynasorb'.



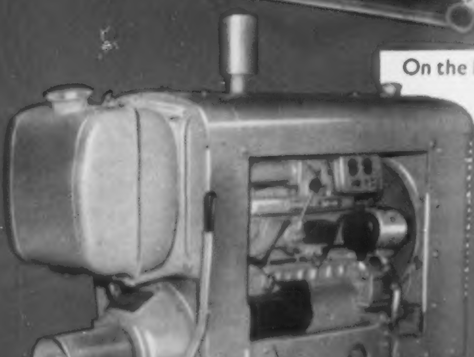
On the ROLLS-ROYCE C6 SFL SUPERCHARGED POWER UNIT



On the DORMAN 6 KA DIESEL POWER PACK



On the LEYLAND DIESEL ENGINE FOR THE OILFIELDS



On the BOLINDER'S STATIONARY DIESEL ENGINES and POWER PACKS

Some users

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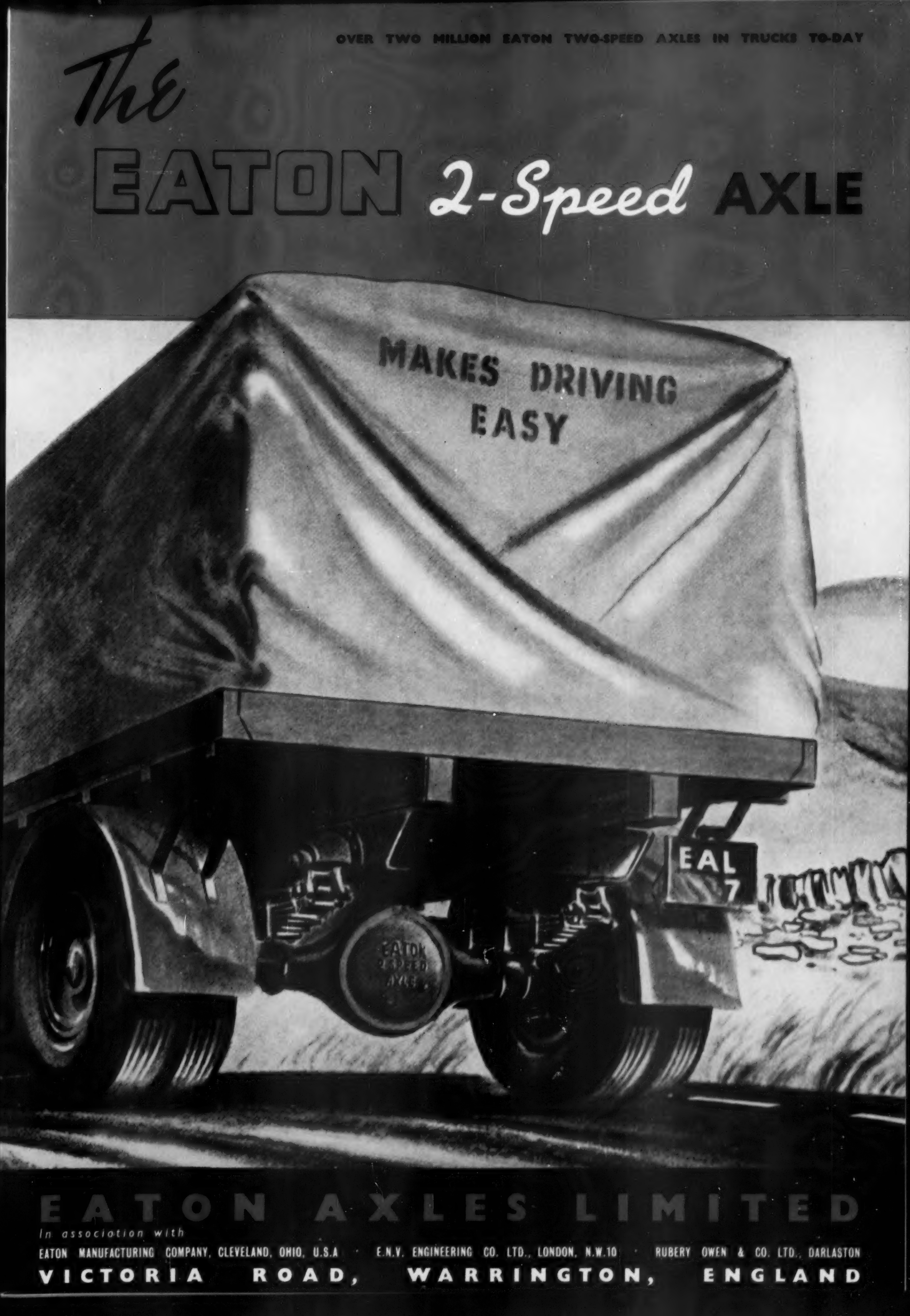
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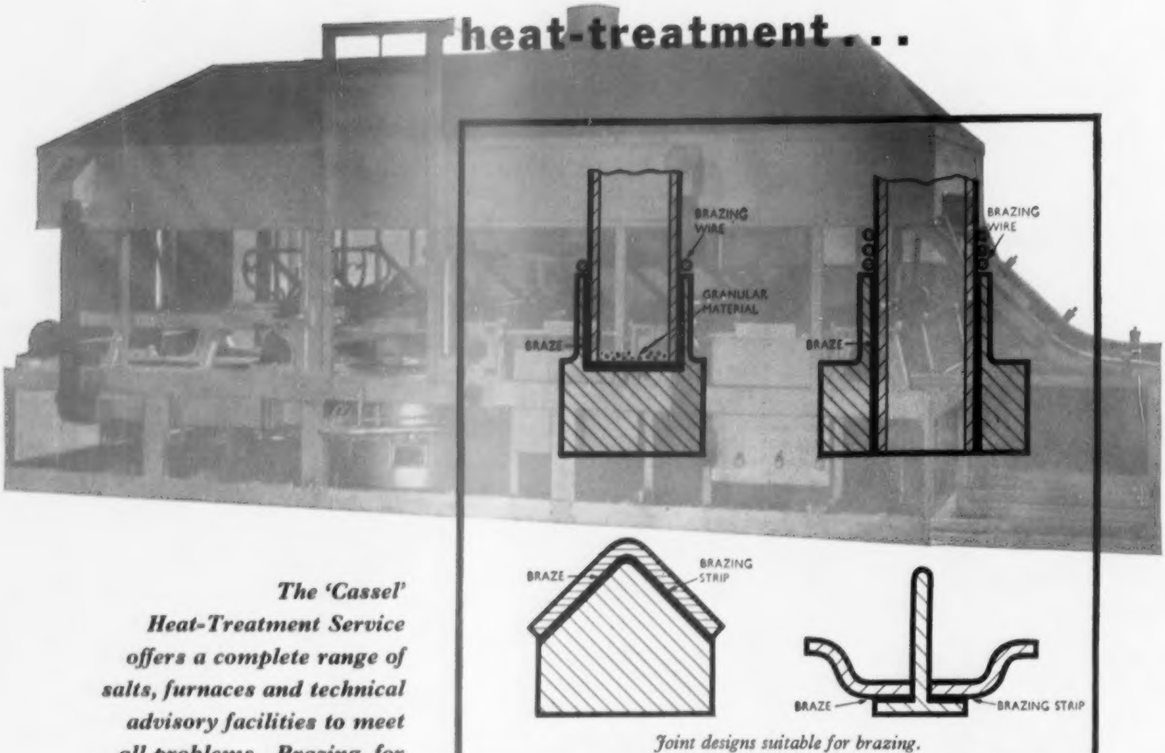

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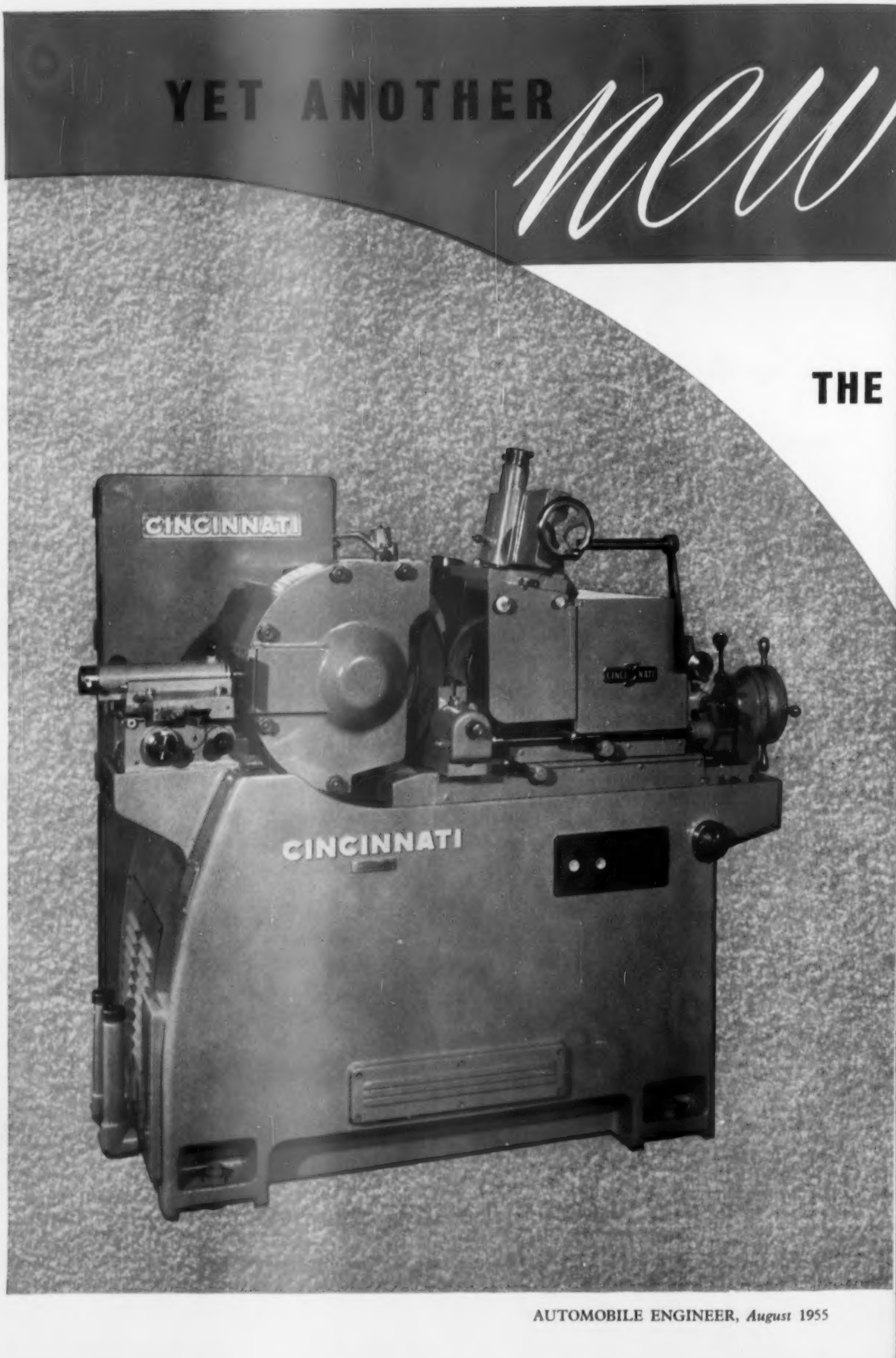
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* *These units are available with either an overdrive or direct top gear*

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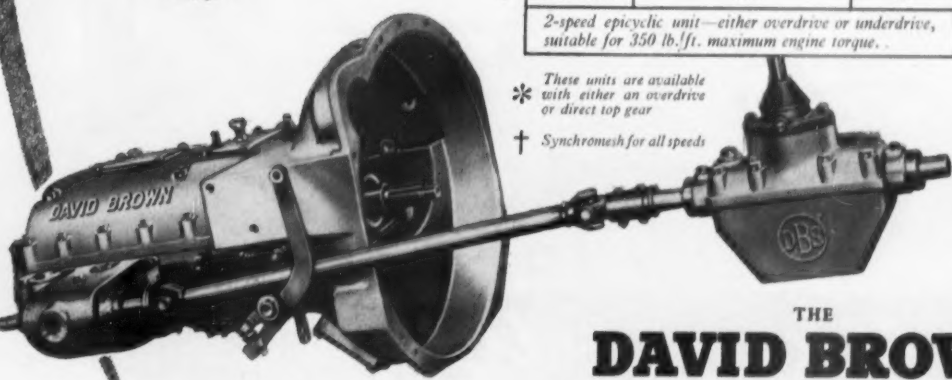

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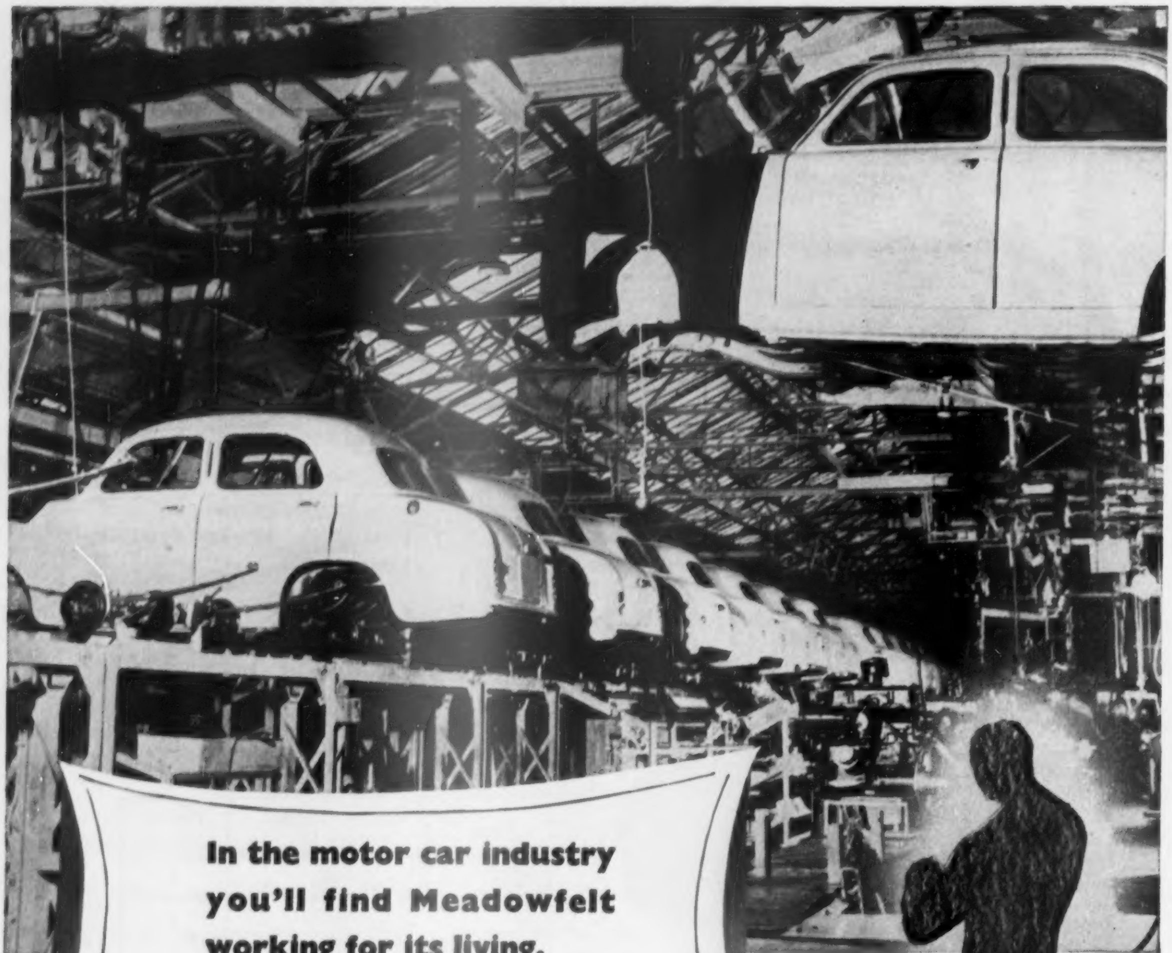

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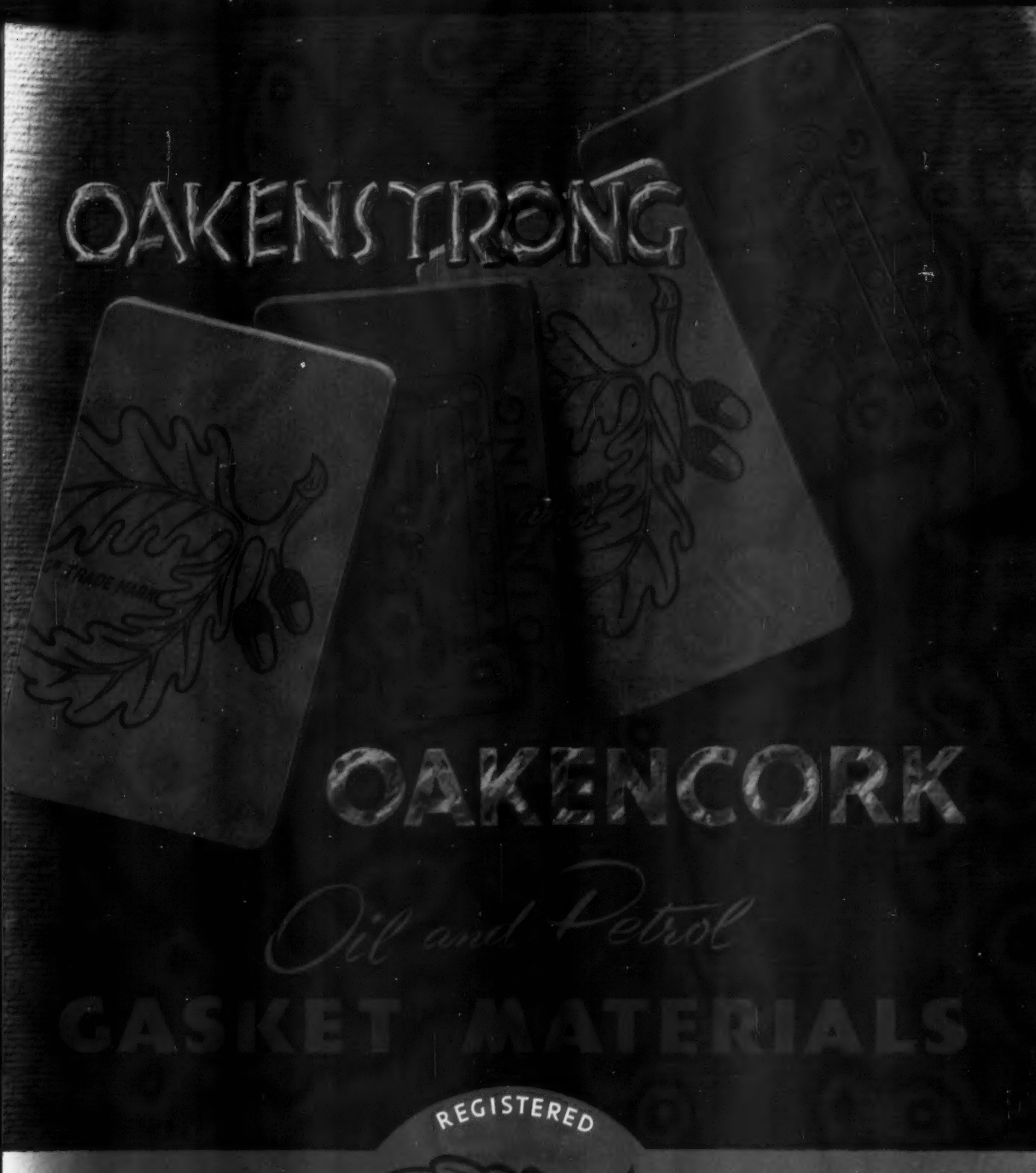
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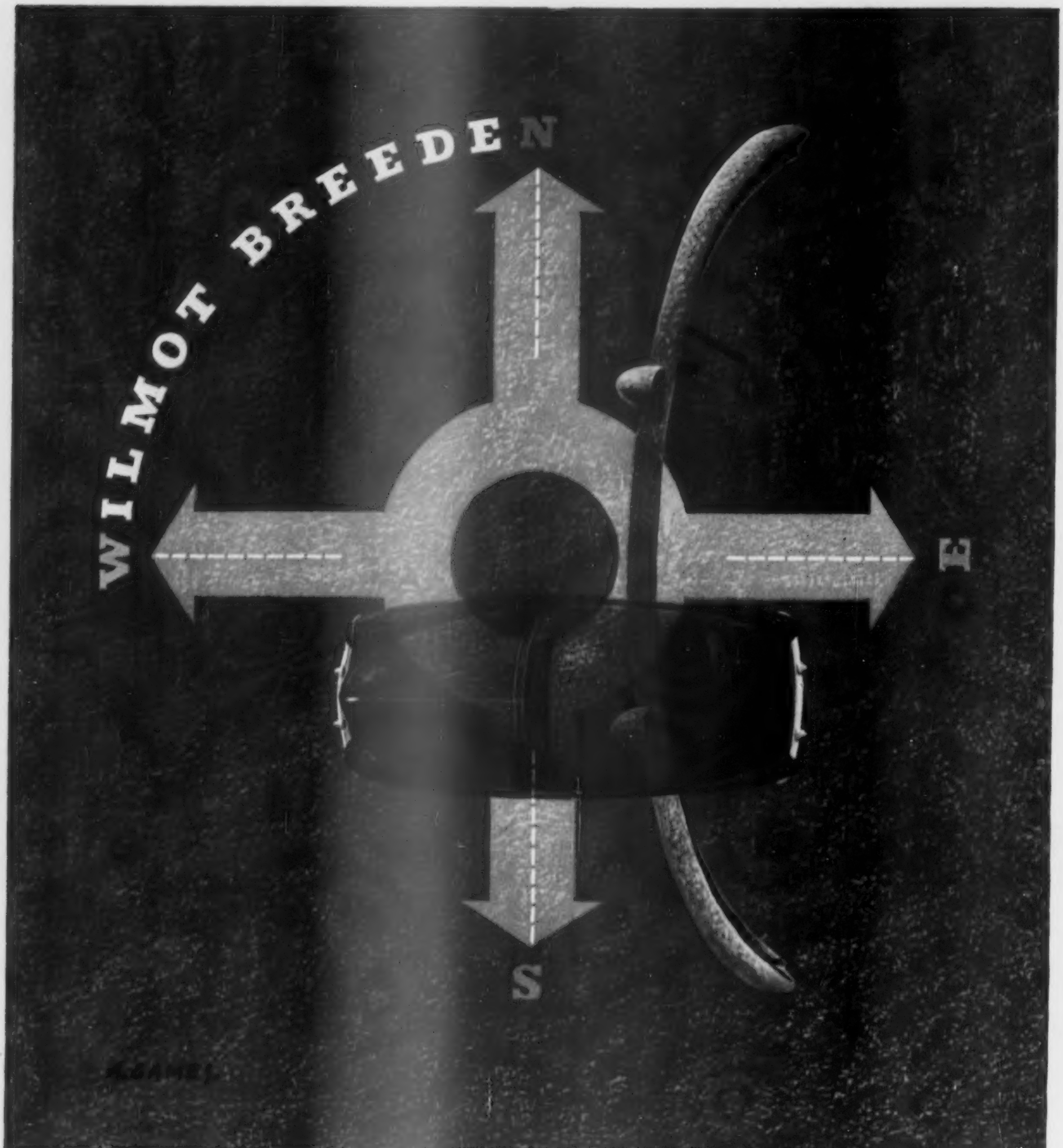
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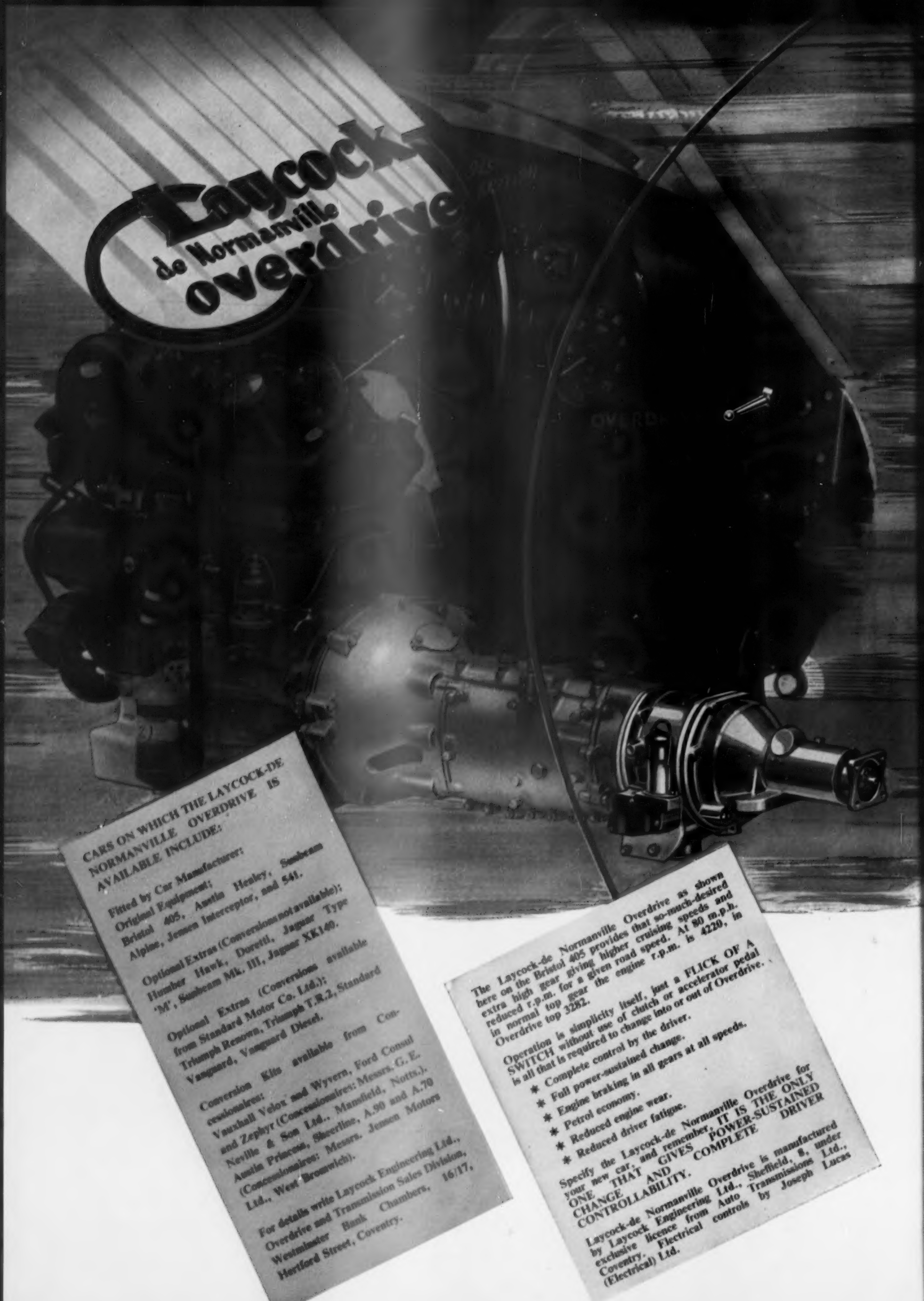
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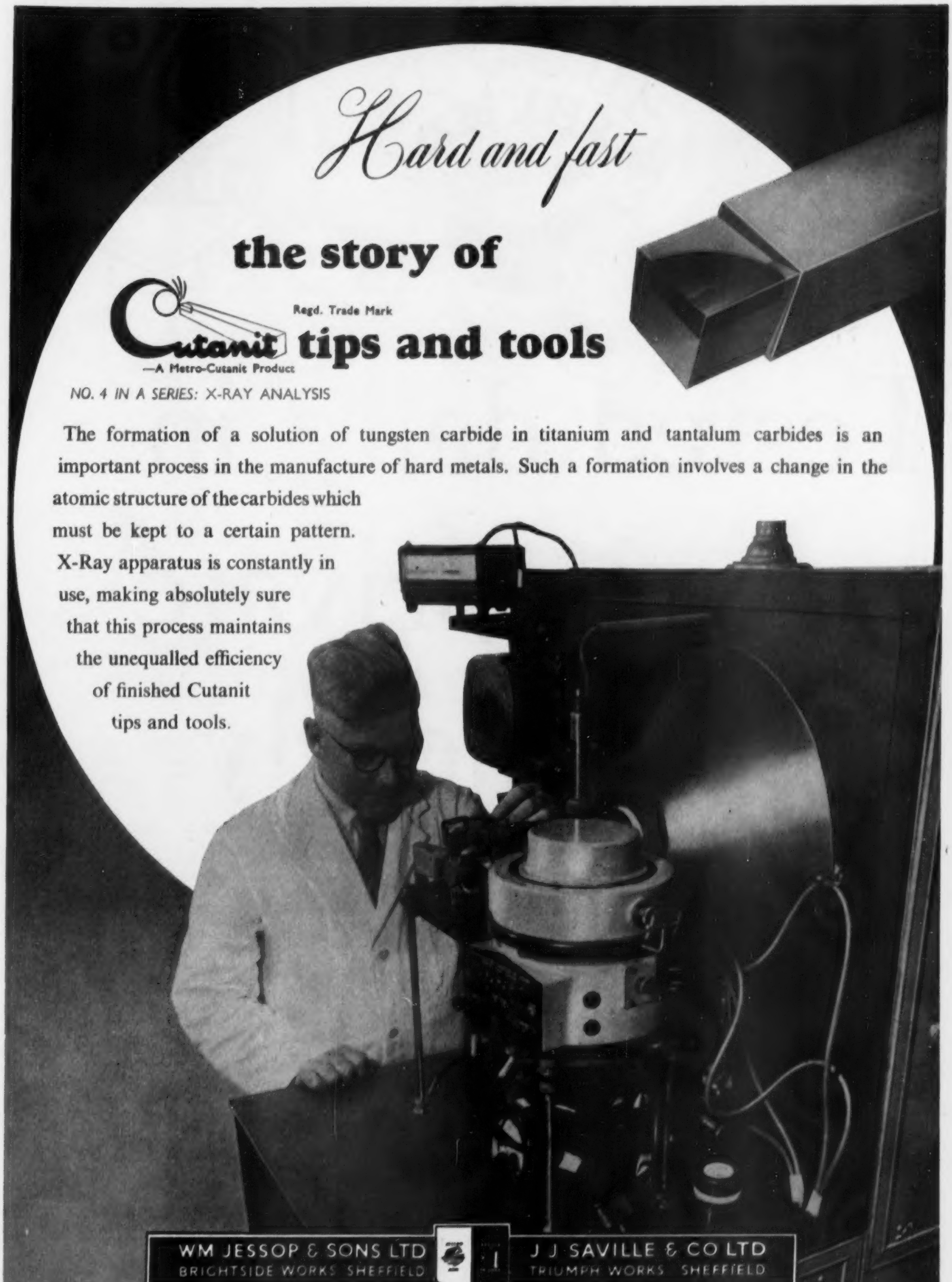
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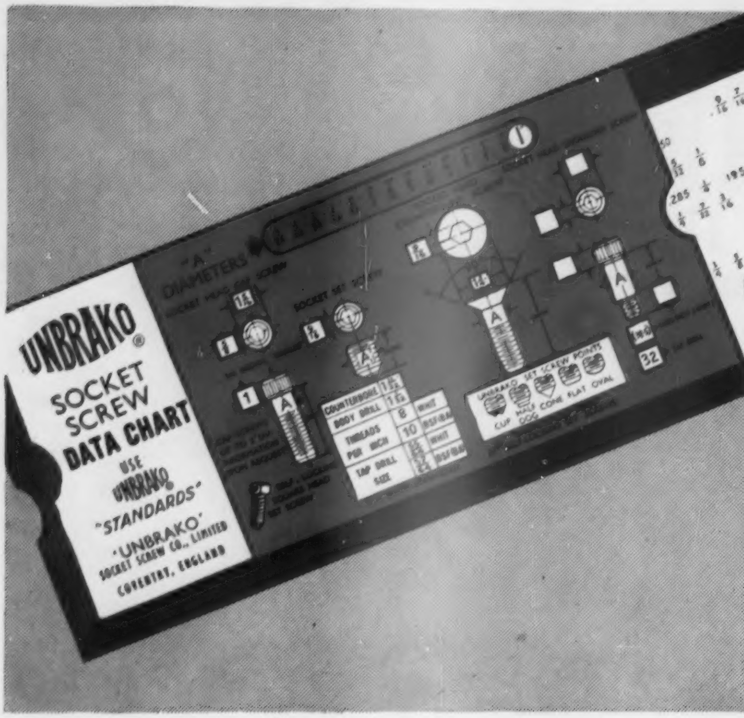

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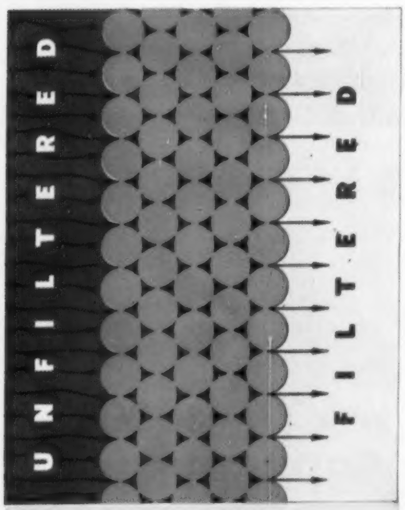

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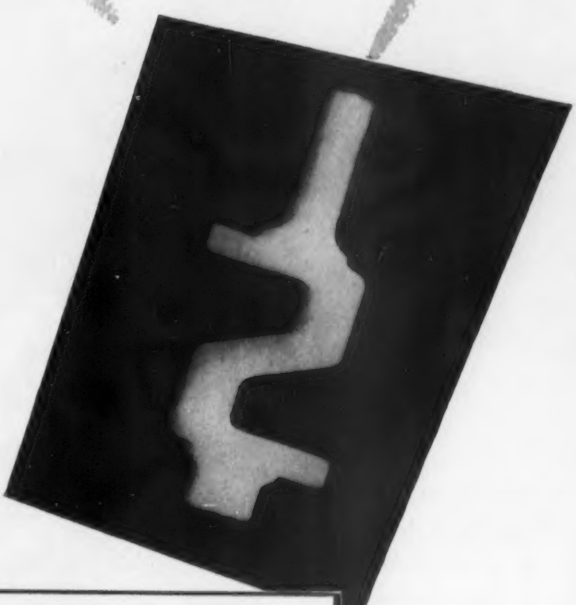

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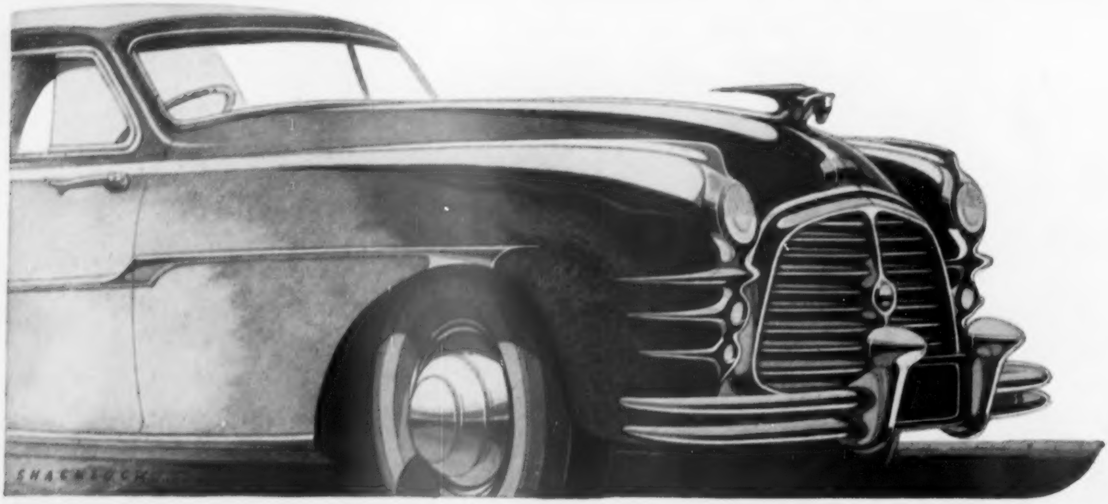

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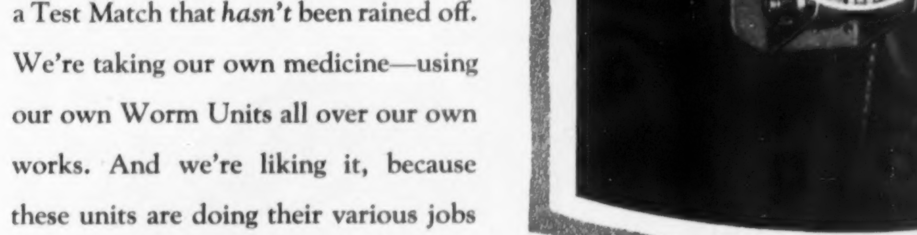

Holroyd

WORM REDUCTION UNITS

JOHN HOLROYD & COMPANY LIMITED · MILNROW · LANCS · TELEPHONE: MILNROW 55393

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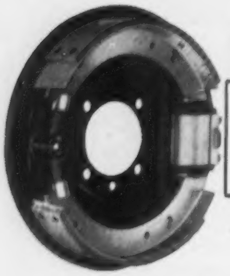
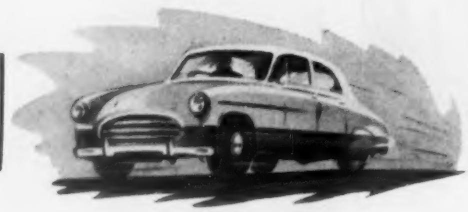
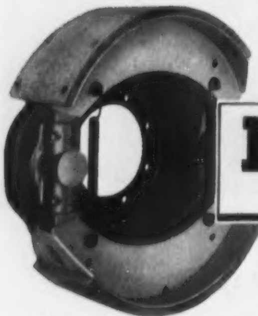


LIGHT
duty



MEDIUM
duty



HEAVY
duty

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THE BEST BRAKES IN THE WORLD

GIRLING LIMITED
KINGS ROAD
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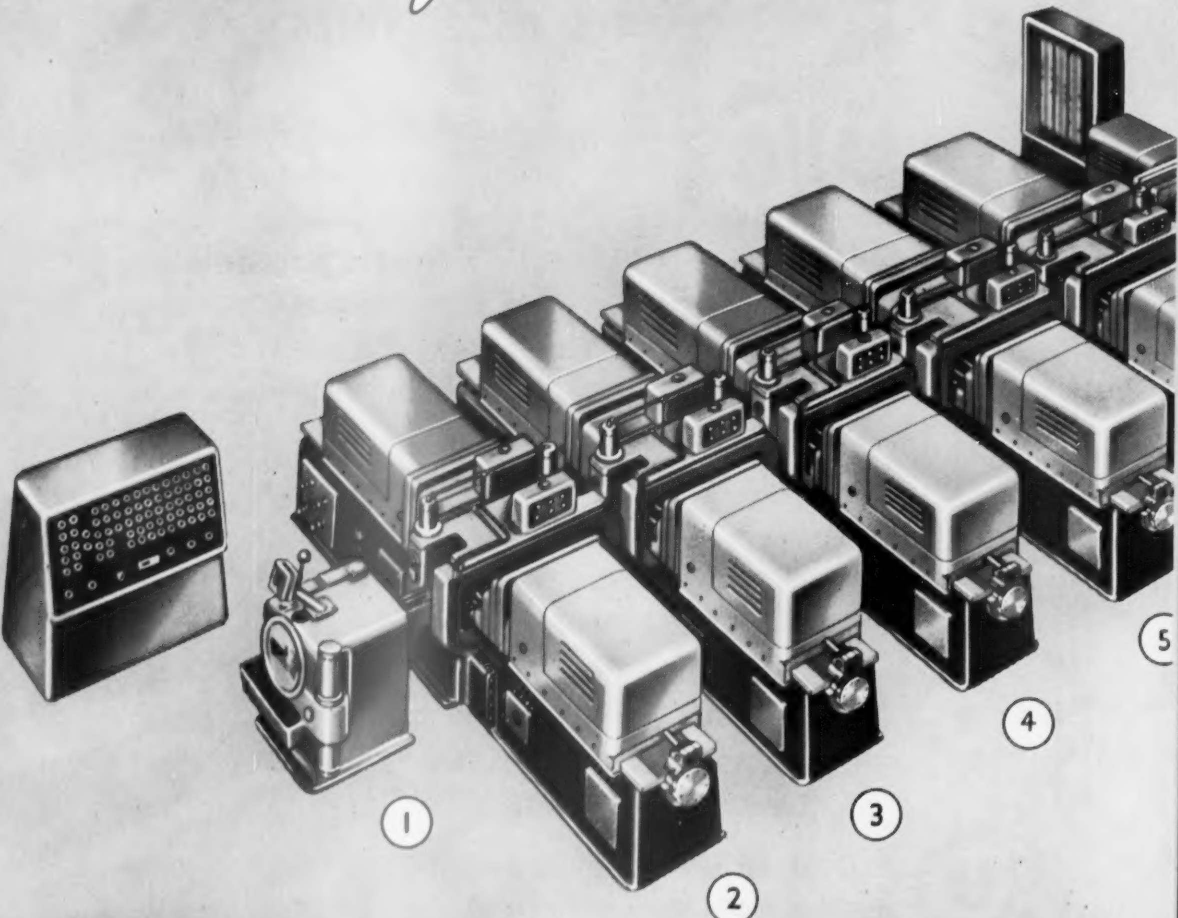
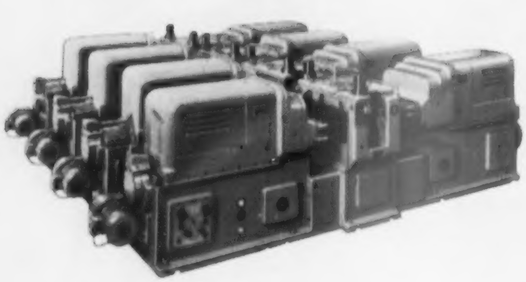
WAY OUT AHEAD

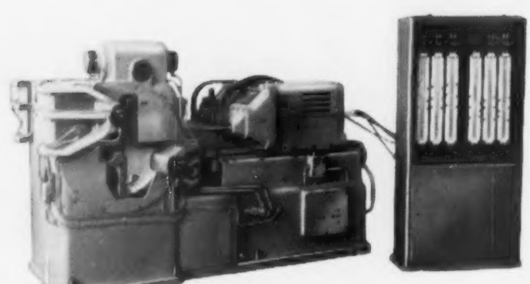
HABERSANG &

Transfer line units



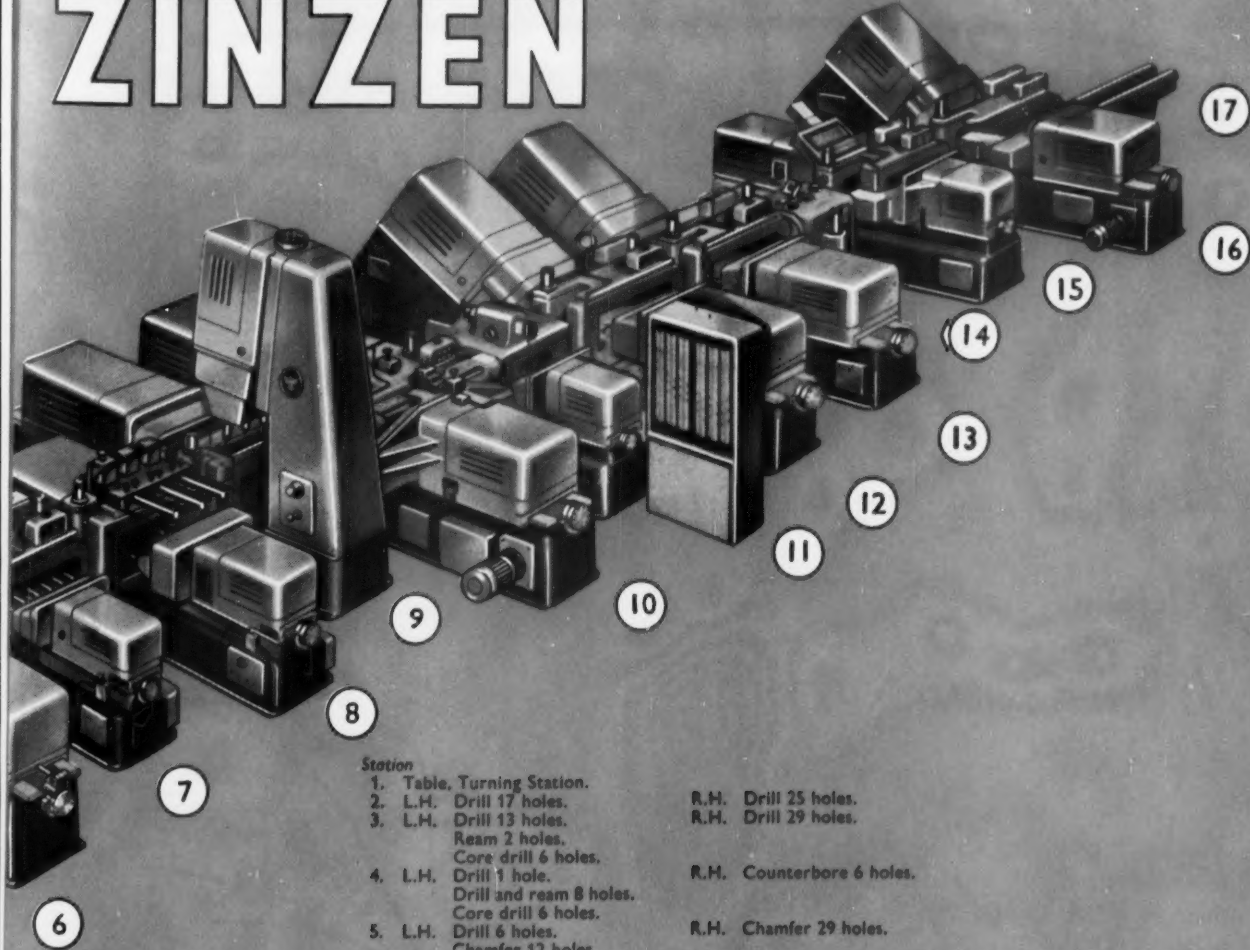


Stations 2, 3, 4 & 5.
Hydraulic units for drilling, reaming and counterboring.



Station 11.
Hydraulic turnover and gauging station.

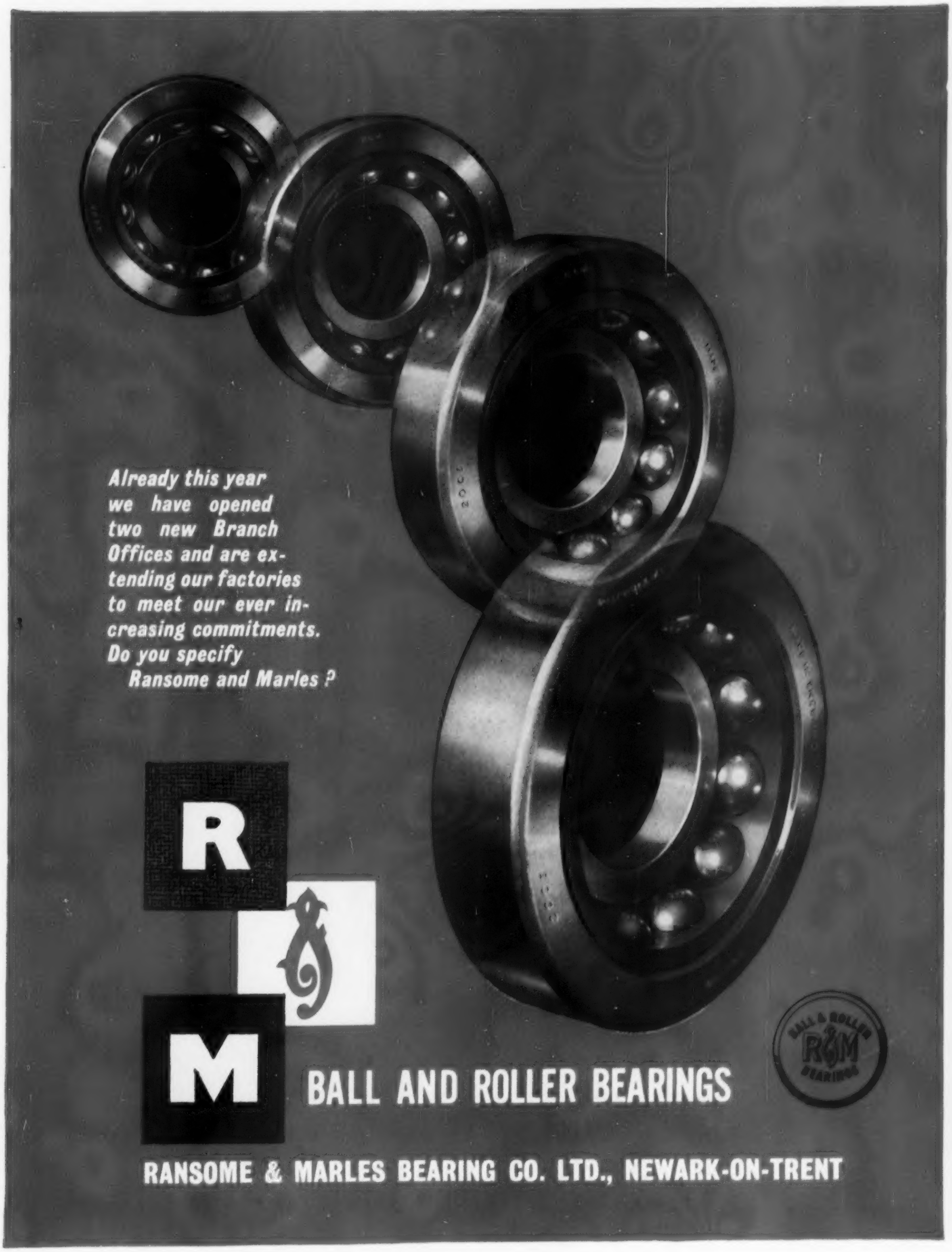
ZINZEN



Station		
1.	Table. Turning Station.	
2.	L.H. Drill 17 holes.	R.H. Drill 25 holes.
3.	L.H. Drill 13 holes. Ream 2 holes. Core drill 6 holes.	R.H. Drill 29 holes.
4.	L.H. Drill 1 hole. Drill and ream 8 holes. Core drill 6 holes.	R.H. Counterbore 6 holes.
5.	L.H. Drill 6 holes. Chamfer 12 holes.	R.H. Chamfer 29 holes.
6.	L.H. Core drill 8 holes.	R.H. Drill 1 hole. Core drill 6 holes.
7.	L.H. Gauge 28 holes.	R.H. Gauge 48 holes.
8.	L.H. Press in 6 valve seats.	R.H. Tap 12 holes.
9.	L.H. Counterbore 12 holes.	*Vertical.* Drill and chamfer 2 holes.
10.	L.H. Ream 12 holes.	R.H. Core drill and ream 1 hole. Drill and chamfer 2 holes. Gauge 12 tapped holes.
11.	L.H. Turning Station.	R.H. Gauge 12 valve guides.
12.	L.H. Drill 6 tapping holes.	R.H. Drill 8 tapping holes. Counterbore 3 holes.
13.	L.H. Counterbore 6 plug seats.	R.H. Chamfer 11 holes.
14.	Hydraulic Turning Station.	
15.	L.H. Tap 2 holes. Gauge 6 holes.	R.H. Gauge 11 holes.
16.	L.H. Tap 6 plug holes.	R.H. Tap 10 holes.
17.	Unloading Station.	

End-to-end time—48 secs. approx. or 75 per hour.

This Habersang & Zinzen transfer line, operating in conjunction with a Heller milling line, is designed to drill, ream, counterbore, and tap 75 engine cylinder blocks an hour. It includes several gauging stations and a station for pressing-in valve seats, and is typical of many arrangements now being supplied, made up from individual Habersang & Zinzen units.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BALL AND ROLLER BEARINGS
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RANSOME & MARLES BEARING CO. LTD., NEWARK-ON-TRENT

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Automatic Lubrication soon pays for itself!

*INCREASES YOUR PAYLOADS 10%

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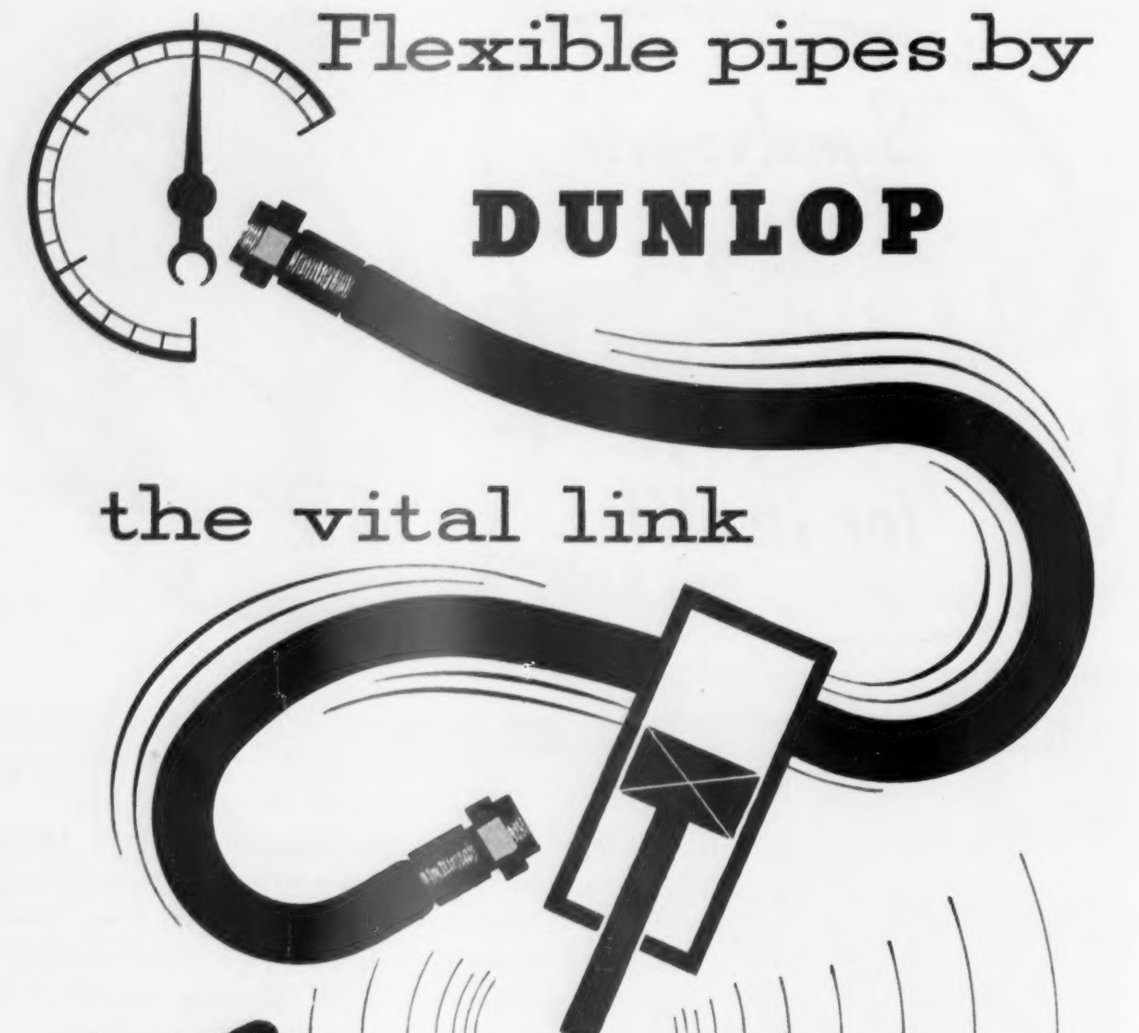

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Conversion Pack for cross flow filter

Conversion Pack for down flow filter

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FUEL INJECTION EQUIPMENT

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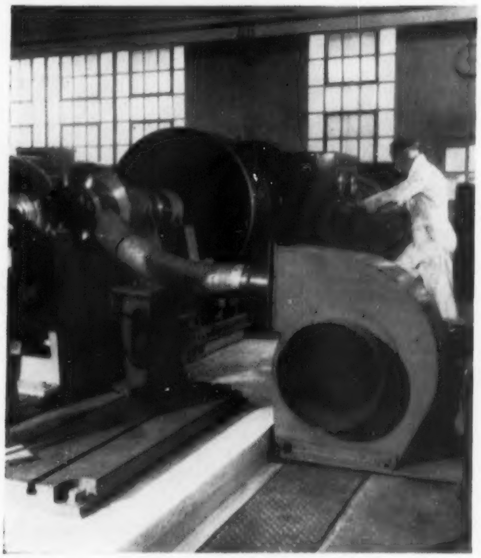
Photograph by courtesy of The Austin Motor Company Limited.

FERODO . . .



A Ferodo technician reads off results as a brake lining undergoes a test far harsher than normal road usage. In the uniquely equipped Test House, experimental linings are tried and proved, as well as linings selected at random from normal production batches. Further tests are made by the powerful cars of the Ferodo Test Fleet both on the road and on the M.I.R.A. track.

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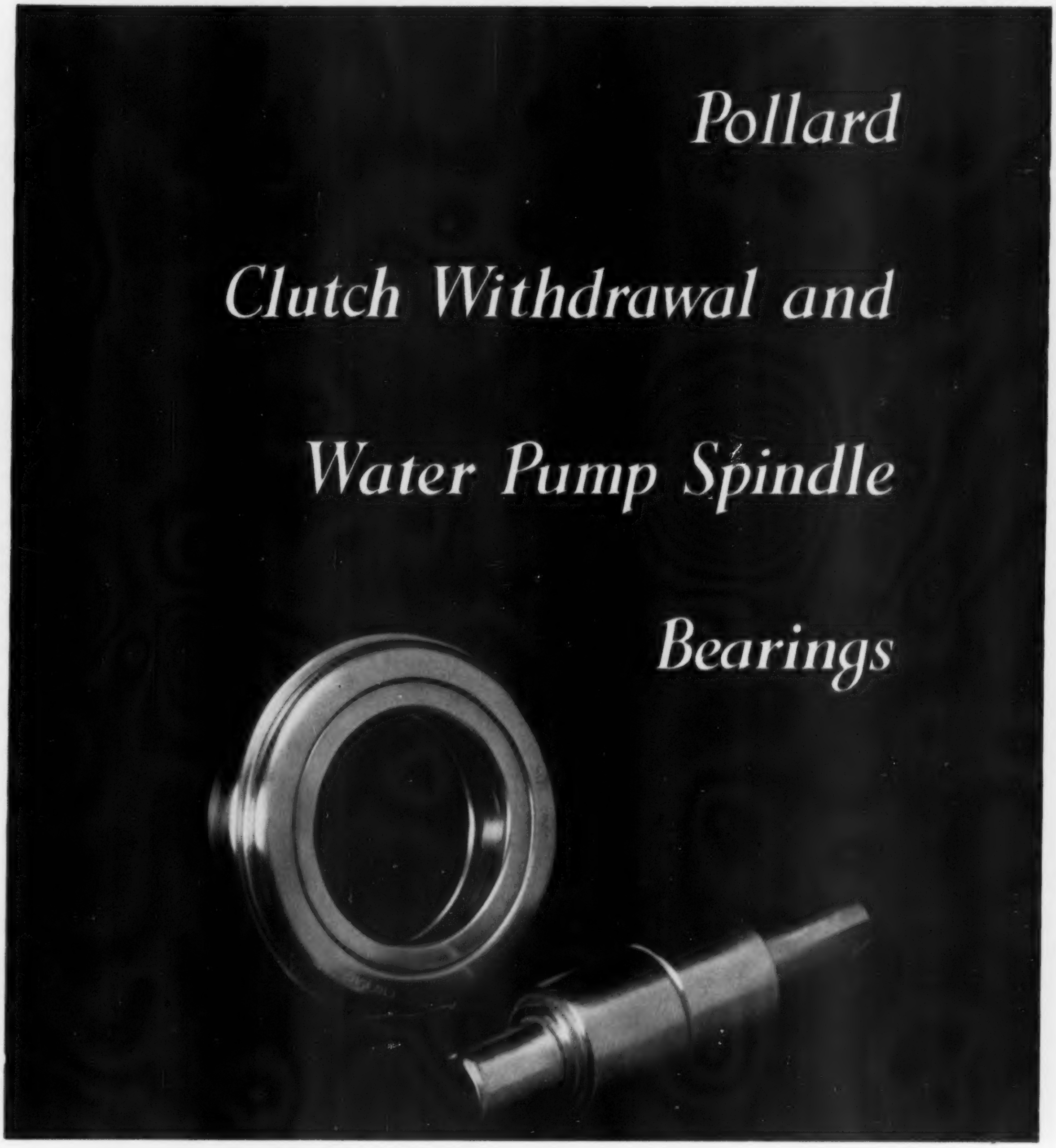
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10th November, 1954.

Dear Sirs,

You will remember that before our 1954 season started you equipped four of our 30ft x 8ft 9 ton Coaches with the Oetiker exhaust brake.

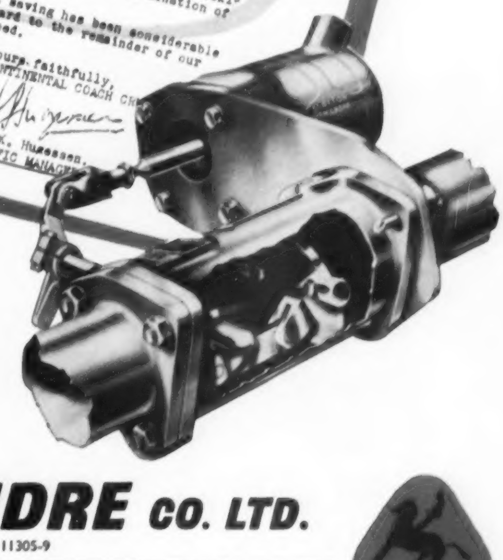
Our season is now over, and we thought you might be interested to have a few details as to how they have performed.

Coach No. MLF.342 covered 24000 miles during the season on our Italian Tour. No brake drums were replaced and on examination of the brake linings we find that 9/16" of the original linings still remain. A similar coach which operated on the same route had two changes of brake linings and one change of brake drums. The itineraries on which these coaches operated took them through seven countries and over the St.Gotthard, Susten, Brunnig and other Passes. Those coaches fitted with the exhaust brake were able to descend the hills more quickly and, of course, much more safely owing to elimination of brake fade.

The financial saving has been considerable and we are now looking forward to the remainder of our fleet being similarly equipped.

Yours faithfully,
for BLUE CARS CONTINENTAL COACH CR[...]

R.J.K. Husessen,
TRAFFIC MANAG[...]



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successful experience with Clayton Oetiker exhaust brakes is shared by other cost-conscious fleet owners

★ You owe it to yourself to investigate this superior braking system. Just 'phone or write us for specific information. Or ask to have a Clayton representative call on you.

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Brake
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for high production repetition work

Archdale 24" PRODUCTION Millers

- AUTOMATIC CYCLES
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Especially designed for high production repetition work, this machine has a good record of performance at The Villiers Engineering Co. Ltd., Wolverhampton, where it is shown form milling gear box selector forks. Table cycles are fully automatic, ensuring the highest possible production on both climb and pendulum milling. Traverse can be extended to suit special requirements.

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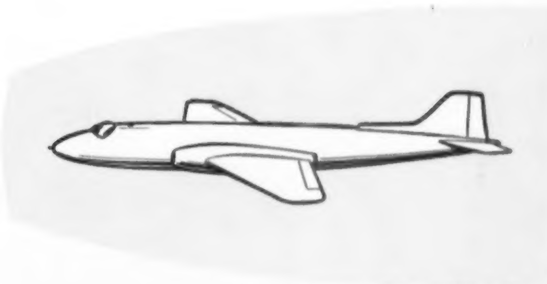
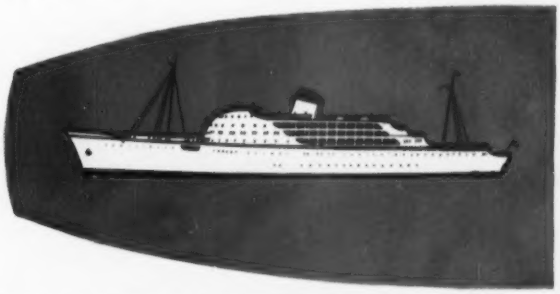
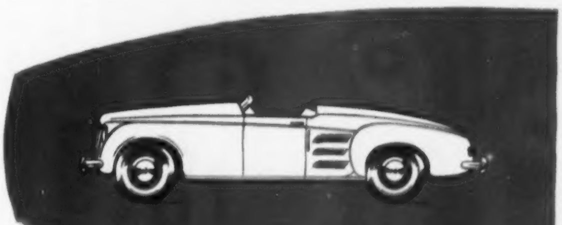
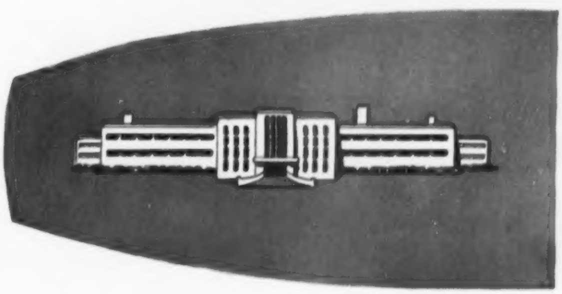
Improved principles of **FUEL BURNING . . .**

the contribution of Lucas research









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Lucas gas turbine equipment is fitted to the majority of British aircraft, including the Comet, Viscount and Britannia. The Olympus engines of the Canberra which attained the altitude record of 63,668 ft. were also equipped with Lucas components.

● Lucas fuel-burning equipment is on continuous ocean service in a number of modern ships.

● Lucas equipment is installed in the engine of the revolutionary new gas-turbine car—first of its type in the world—made by the Rover Co. Ltd.

● Lucas equipment is incorporated in the gas turbine engines used to power generating plant in large marine and industrial installations.

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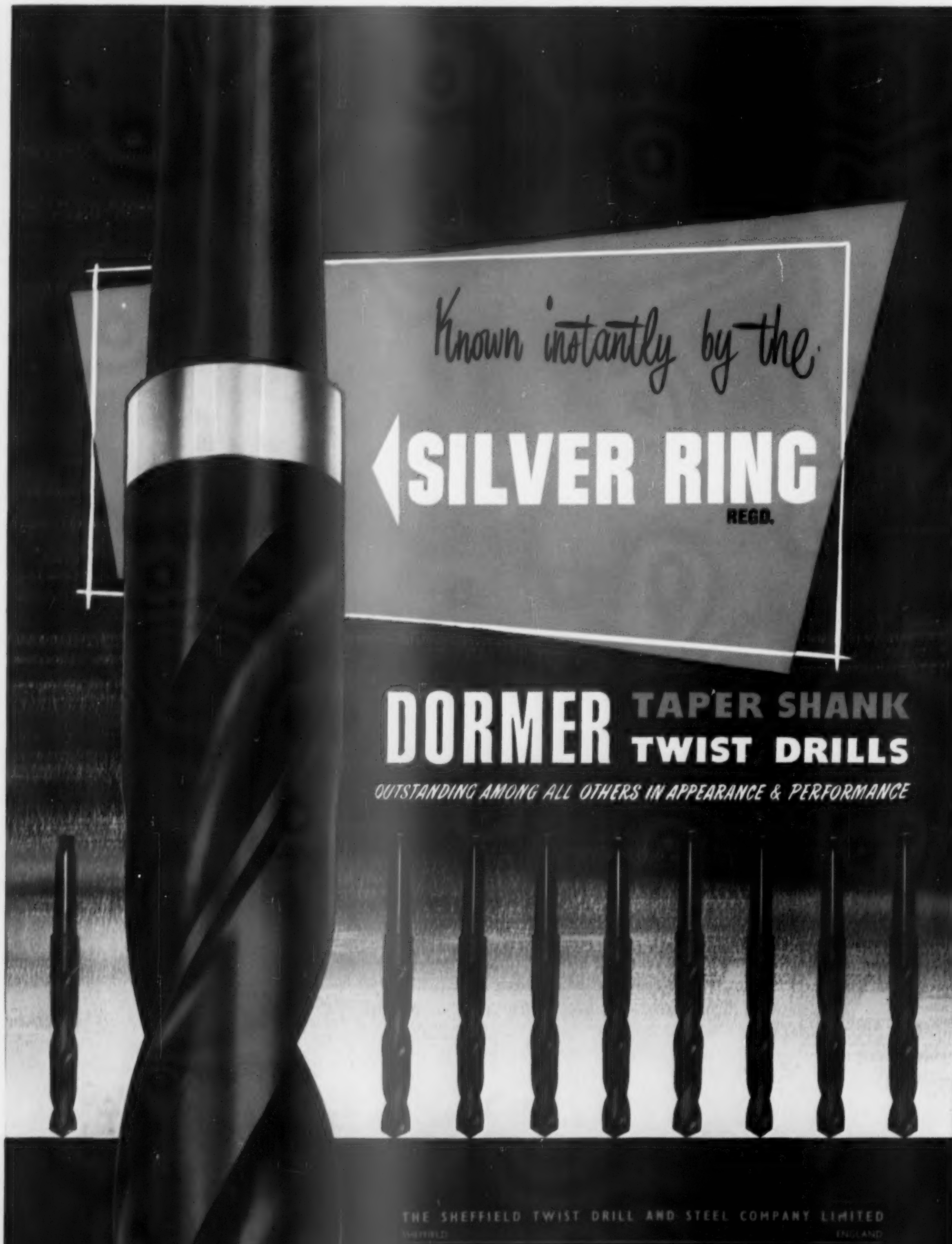

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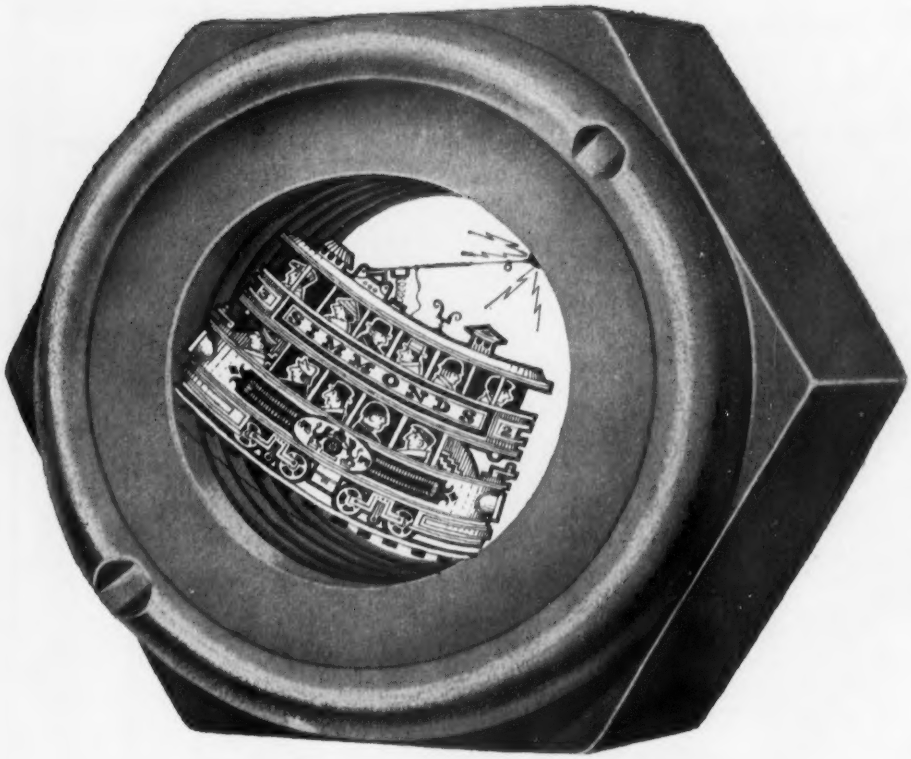

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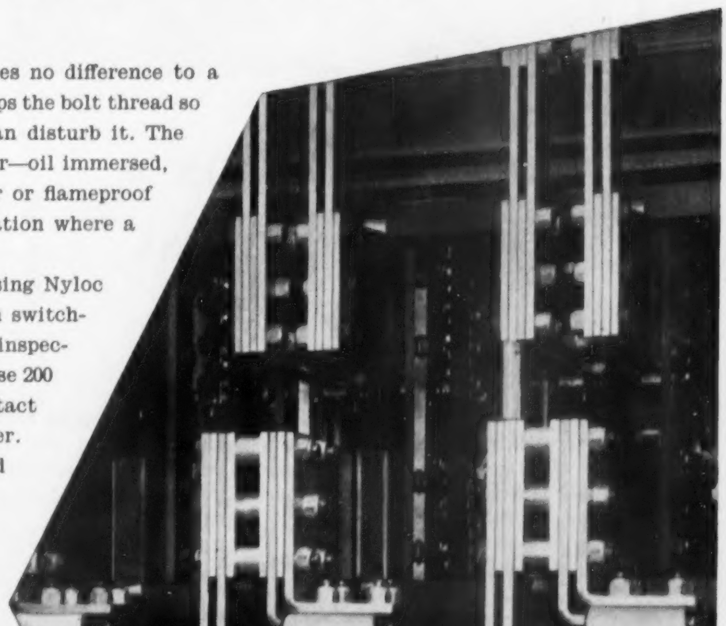

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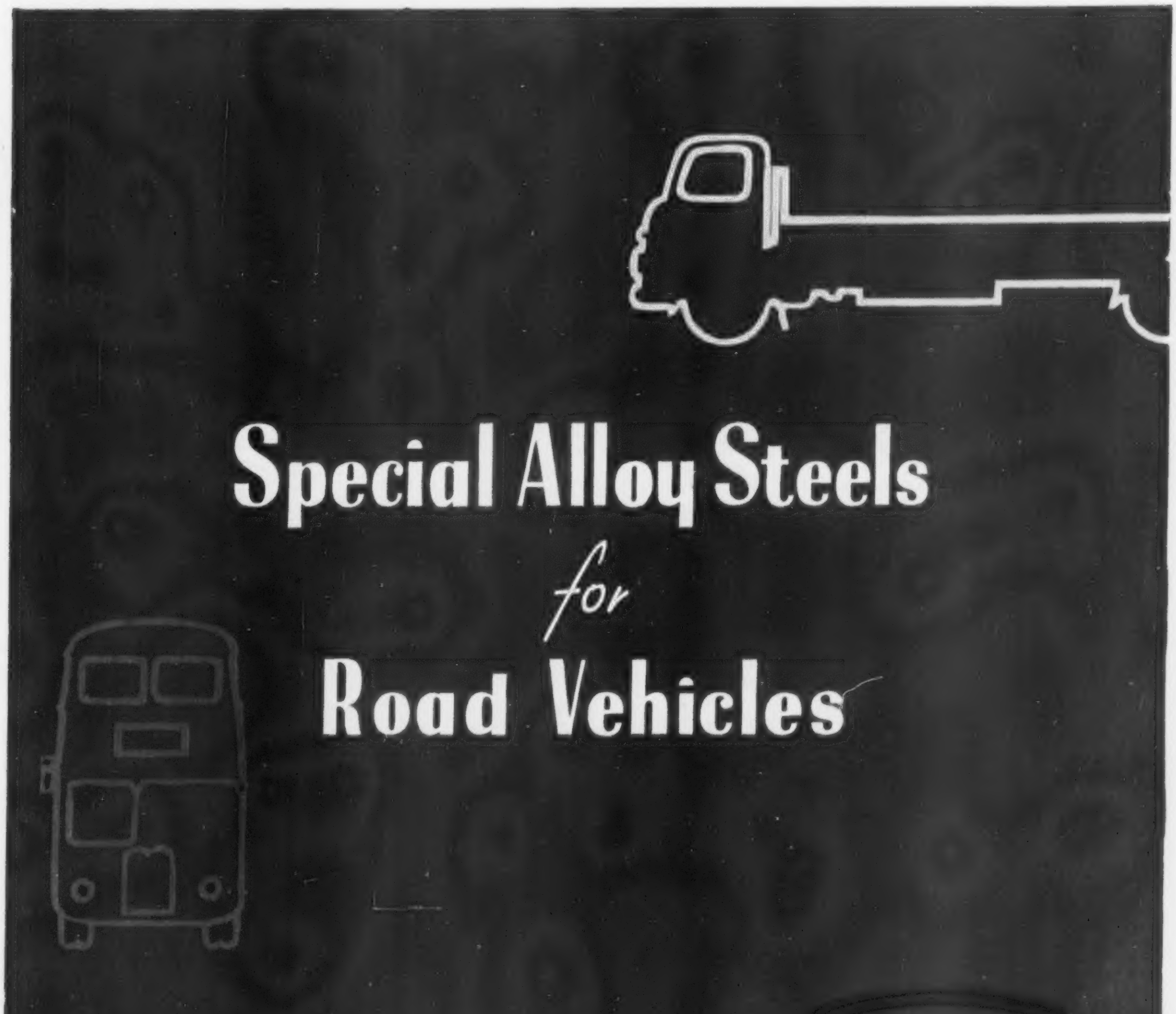
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Design, Materials, Production Methods, and Works Equipment

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Publishers: ILIFFE & SONS LTD., DORSET HOUSE, STAMFORD STREET, LONDON, S.E.1.
Telegrams: Sliderule, Sedist London **Telephone: Waterloo 3333 (60 lines)**

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PUBLISHED MONTHLY—SECOND WEDNESDAY. ESTABLISHED 1910

Annual Subscription: Home and Overseas £2 11s. 6d., including the Special Number; Canada $7.50; U.S.A. $8.00

VOL. 45 No. 8 AUGUST, 1955 PRICE 3s. 6D.

Technological Education

IT has long been recognized that this country has lagged far behind the United States of America and Germany in the provision made for educating and training technologists. Modern industrial developments all tend to reduce the number of men required on the shop floor, but at the same time they increase the need for highly trained technologists. Furthermore, developments arising from fundamental research are frequently made suitable for industrial application only after technologists have worked on them.

That there is a problem is now officially recognized. Its magnitude can hardly be over-stated. For example, the British Transport Commission say that they will require 500 electronic mechanical engineers in carrying out their reorganization plans. In some quarters, this is regarded as an understatement if full advantage is to be taken of new techniques and devices.

As a first step, the Minister of Education is setting up a council for the purpose of introducing and administering a new national award in technology. This council, which will be autonomous, will have the duties of approving the syllabuses and conditions of teaching, the qualifications of teachers, the examinations, and the grants of awards to students who satisfy the examiners at the various regional technological colleges, which are yet to be set up. Lord Hives has accepted the chairmanship of this council; this may be taken as a guarantee that very high standards will be called for.

Apparently, it is the intention that the courses in such subjects as mathematics, science and engineering at the regional colleges should not differ materially from those of the universities. The award, probably to be known as the Diploma of Technology, will, as far as we can gather, be based on a standard equal to a university pass degree, but for a narrower range of subjects. It seems reasonable to assume that the diploma will rank between a university degree and a Higher National Certificate.

There still remains much to be done before these embryo plans are translated into an accomplished fact. Already there are divergences of opinion between the Minister of Education and the professional institutions, which, of course, have a very close interest in this matter. The plans, so far as they have been disclosed, suggest that the official intention is to set up a number of regional colleges, each dealing with a single technology; this does not meet with anything like universal approval.

In a joint letter to the *Times*, the Presidents of the Institutions of Electrical, Mechanical and Civil Engineers say:—"The policy upon which the Minister has decided differs materially from that which our Institutions have advocated in discussions with him and his predecessors during recent years. Our Councils have expressed the conviction that the country must make the most effective use of the limited number of teachers available to give technological education of the quality required. Furthermore, the money for the necessary physical facilities would be applied far more effectively if it were concentrated in relatively few places."

"In the view of our Councils it would be necessary at the outset to concentrate national resources within a small number of colleges of technology chosen on the basis of the comprehensiveness of their technological facilities and potentialities, and not on a capacity to deal with a single technology only. These colleges would need to have a large measure of academic and financial autonomy. These views are widely held within the engineering industry and also by other professional institutions and societies."

At the outset, the difficulty will be to obtain teaching staff of the requisite quality. In the main, the staff must be recruited from university graduates, and the colleges will have to face competition from two sources, the universities and industry. Appointment to a university post will carry a cachet which, at least in the early days, will not apply to technological college appointments. An industrial career can offer possibilities of much more glittering prizes than the teaching profession. If the right people are to be attracted, there must be some radical re-thinking about the financial rewards attaching to teaching appointments.

Plastics Press Tools

EXPERIMENTS have been carried out in this country in making press tools from plastics materials. They have not met with any great measure of success. Recent developments in America suggest that further investigations into the possibility of using plastics material are warranted. In the United States, the Budd Company of Philadelphia—the largest independent producer of automobile pressings in the world—is now making considerable use of reinforced plastics for dies, jigs and fixtures.

It is not suggested that such dies are suitable to replace conventional metal dies for large quantity production, although their life is much greater than might be expected.

For example, a complete die-set—punch, blank holder and lower die—has been used to produce over 6,000 inner quarter panels for a hard top convertible. This suggests they would be suitable for quite a number of applications in this country.

Even for large quantity production, the Budd Company often starts with tools made from reinforced plastics. A large metal die made by conventional methods may take from six to eight weeks to complete, whereas a die of reinforced plastics can be made in about six weeks with the pattern as a mould. It is therefore the practice to make reinforced plastics tools and try them out. Any modifications, and large die sets usually need some, can be made much more quickly and easily on the plastics tools than they could be on metal tools. The tools are then put into production and are used until the output obtained from them, justifies their cost. Meanwhile, the manufacture of the conventional tools is carried out. It does appear that this practice might with advantage be carried out in this country also.

Research

WITHOUT departing in the slightest from our belief that the Motor Industry Research Association has earned and should have the utmost possible support from all sections of the automobile industry, we heartily welcome the opening by the Ford Motor Company Limited of a Research Centre in Birmingham. Although this Centre is intended to deal only with relatively long term investigations, its activities should be complementary to, and in no way competitive with, those carried out in the M.I.R.A. laboratories.

Since long term applied research is the reason for developing the Centre, the decision to site it in the Midlands rather than in the Dagenham area is wise. Experience has shown that only too often does it occur that a research department is called upon more and more to deal with urgent day-to-day problems, until finally, trouble-shooting and development rather than research are the actual functions of the department. The distance between Birmingham and Dagenham, should, in itself, prove an effective barrier against this happening at the Ford Research Centre.

There are, of course, other reasons for the choice of a Midland site. Despite the great degree of integration that has been effected in the Ford plants, Ford cars still incorporate much material and many components obtained from independent sources, and most of the main suppliers operate within easy reach of Birmingham. Liaison and personal consultations between the technologists employed at the Centre and suppliers' technologists will be important and will be more easily maintained than would have been possible if the new laboratories had been sited in the Dagenham area. The proximity of the M.I.R.A. proving ground is another great advantage. We have no doubt that the fullest possible use will be made of the splendid facilities at Lindley to supplement and confirm the results of laboratory tests.

The segregation of the research laboratories does, however, hold one danger; a tendency may develop to pursue knowledge for its own sake—often a highly laudable form of activity, but far from the *raison d'être* for the Ford Research Centre. In all research work the aphorism "the good is the enemy of the best," must always be kept in mind, but in applied research it is equally important to remember that there does come a time when seeking further refinement is wasteful. It is therefore important that those controlling the Centre should always realize they are employed to find means of improving motor cars, either by making them more efficient or by making equally efficient vehicles at lower cost. The work at the Centre would probably benefit through seconding key members of the staff for short tours of duty with the Design Engineering Division at Dagenham. It might, in fact, be good policy to carry out seconding on a reciprocal basis.

CONTENTS

SAURER TYPE 4GP BUS

A Vehicle in Which the Engine is Installed in an Articulated Passenger-Carrying Trailer

PUBLIC service vehicles must be designed to meet not only the requirements for reduced operating expenses but also to occupy as little road space as is practicable. It is in this respect that the main difference between British and foreign operation becomes apparent, that is, in the strictly and narrowly limited number of standing passengers on the one hand, and the virtually unlimited numbers on the other. An eminent, Continental vehicle designer travelling recently in a London bus expressed utter amazement at the conductor's refusing to take standing passengers, and was heard to say "He is throwing money away." When the no-standing rule was explained to him, he pondered a long while, and in the end exclaimed that he could not believe it: the British, he pointed out, were both clever and practical and this rule was obviously introduced as one of their hidden reserves. It would seem that the time is near when the possibility of using these hidden reserves might have to be considered.

The employment of double-deck vehicles has never been greatly favoured either on the Continent or in the United States of America, the chief exceptions being found with buses in Berlin and Hamburg, while a few of these vehicles have been operated in the United States, Denmark and Spain. Previously, double-deck tramcars were tried out in Austria, Denmark, France and America, but were soon abandoned because of their relatively low overall speed of operation due to time lost at stops. Generally, they were reasonably successful so long as they operated on what were considered abroad as luxury lines, that is, practically all passengers seated with perhaps up to half a dozen standing ones. Normal single-decker practice overseas is that only about 40 per cent of the passengers carried are seated.

An obvious disadvantage that has to be faced with double-deckers is a high wage bill, since a crew of two is required to handle about sixty passengers, but against this the claim is often made that they occupy a smaller space on the road, and thus reduce congestion. However, a closer analysis reveals that this claim is scarcely justified. In the case of the new London Transport, RM 64-passenger double-deck bus, a road area of 340 ft^2 is required. To move 192 passengers would require three of these buses and a crew of six. At stops these vehicles would occupy a road length of $27+4+27+4+27$ ft $= 89$ ft, and the time during which this distance is occupied is relatively long because of passengers negotiating stairs and also because of the restricted entrance and exit facilities. With the new 53 ft long, Zurich, articulated bus, occupying a road area of 435 ft^2, up to 180 passengers can be dealt with by a crew of only two. That there is a potential saving in fuel consumption is indicated by the fact that the frontal area of the Swiss vehicle amounts to 76 ft^2, as compared with $3 \times 113 = 339$ ft^2 for three London buses. In addition, the all-up weights are 25 tons and 3×11 tons respectively.

The development of large capacity single-deck buses has followed on lines similar to the early development of tramcars in different parts of the world. When the capacity of these vehicles failed to meet traffic requirements, trailers were added; but their use made it impossible to meet the demand for higher schedule speeds. Because of this, articulated tramcars appeared in Boston, Baltimore, Chicago, Cleveland, Montreal, and more recently in Calcutta, Duisburg, Dresden, Leipzig, Berlin, Frankfurt, Amsterdam, Dortmund and a number of Italian towns. Most of these vehicles are operated by one driver and two conductors, their main drawback being the time that is lost in the event of derailment. However, no such drawbacks exist with road vehicles,

Fig. 1. The Saurer type 4GP bus negotiating a sharp turn into a relatively narrow road

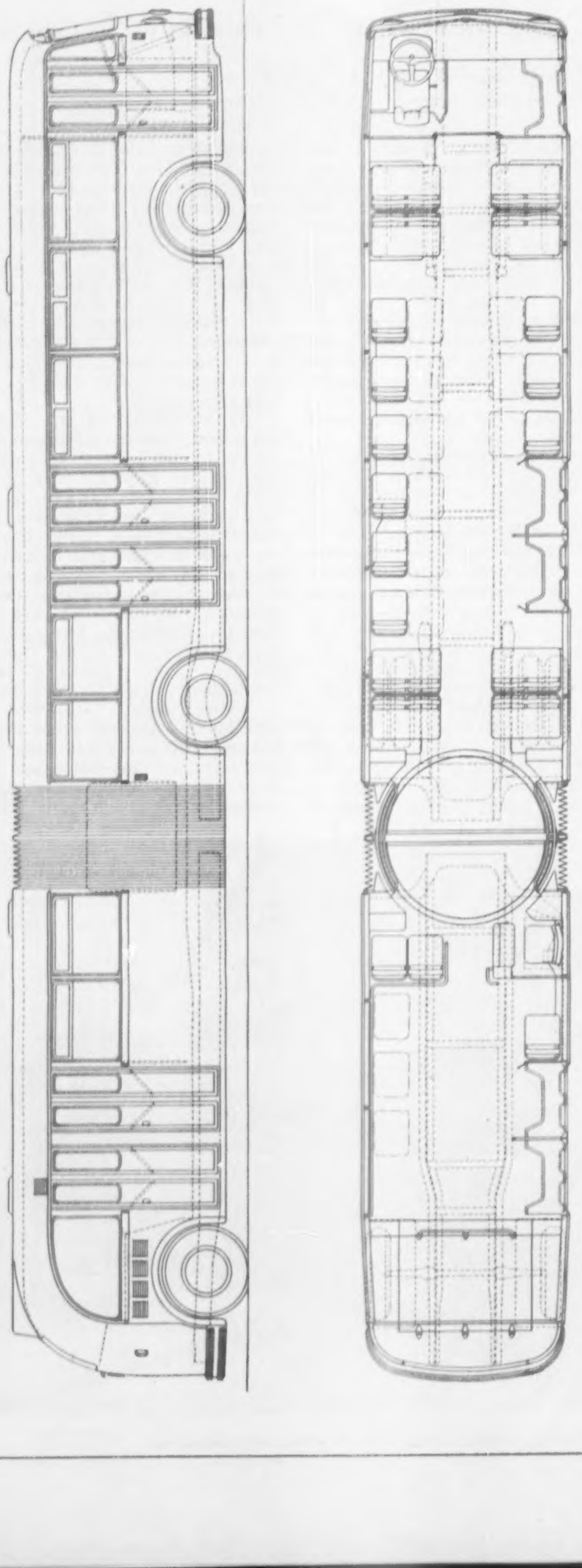


Fig. 2. Arrangement of the Saurer 4GP bus, showing alternative seating layouts. The overall length of the vehicle is 16·180 m, the wheel-base of the front portion is 5·8 m, while the distance between the centre of the rear wheel of the tractor and that of the trailer is 5·15 m

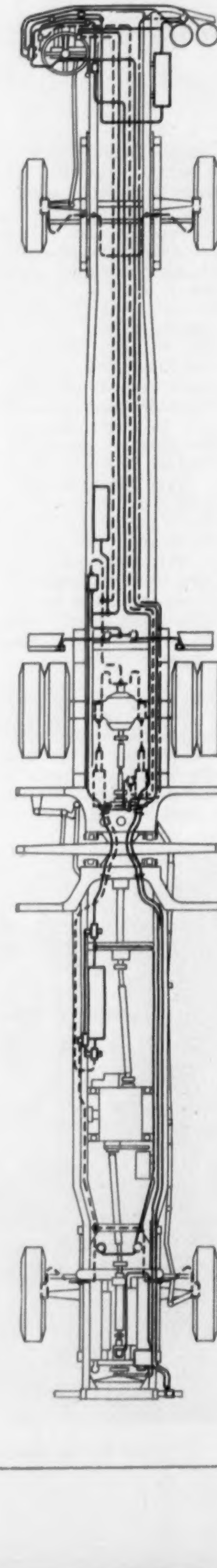


Fig. 3. Layout of the pneumatic system on the chassis of the Saurer 4GP, semi-trailer bus

—— High pressure air system — — — Pneumatic brake system — · — Hydraulic system for releasing the handbrake in the event of the air pressure being too low

and, in 1948, articulated buses appeared in America. These were the 58D Super-Twin-Coaches, with two 140 h.p. engines and carrying 120 passengers. Similar types of vehicle were put into service in Italy, and more recently in Germany, where articulated eight-wheel buses and trolley buses were acquired by the Dortmund and the Neuss municipalities, respectively.

In Switzerland, the traffic requirements are difficult to meet because of the universally adopted two-hour midday break. Most office and factory workers go home for lunch—a practice that is facilitated by the relatively small size of the towns—so four peak traffic periods must be dealt with instead of two. On the other hand, although there are stage fares, the number of stages rarely exceeds three and most regular passengers are season ticket holders or buy books of tickets at reduced price. This facilitates passenger handling on the pay-as-you-pass principle. Most modern Swiss vehicles are of the Peter Witt type, with rear entrance and a centre as well as front exit, the conductor being seated near the rear entrance. Season ticket holders are allowed also to enter at the front, where they show their tickets to the driver. Based on this system, modern Swiss trams and buses are built to accommodate a total of 100 passengers, of whom only about a quarter are seated. This practice is generally accepted throughout the country, particularly in view of the fact that scarcely more than 30 minutes are required to cover even the longest route.

However, recent housing extensions in the northern suburb of Schwamendingen and the district adjoining the main airport at Kloten, together with the already high and ever increasing wages of the crews—which already represent 75 per cent of the operating costs — have made it desirable to provide vehicles of even greater capacity. Therefore, after extensive study of articulated vehicles—notably as used in Italy—the Zurich Municipal Transport (V.B.Z.) in February 1954 placed a contract with Adolphe Saurer, of Arbon, for the supply of the first Swiss articulated bus, Fig. 1, which was put into service on January 17th of this year. This vehicle accommodates 30 seated and 120 standing passengers, Fig. 2. The number of standing passengers is increased to 150 during rush hours, and the vehicle is probably the largest passenger-carrying road unit in existence. Considering the many novel features incorporated in the design, the vehicle has been remarkably successful, having remained continuously in service since its introduction.

Fig. 4. Access to the engine can be gained from the rear quarters

This bus is basically a four-wheel vehicle with normal steering, axles and brakes, attached to which is a two-wheel trailer. The trailer carries the engine and the transmission, which drives the rear axle of the four-wheel unit. To ensure good steering qualities, both the front and rear wheels are steered. The steering of the rear wheels is effected by a linkage actuated by the relative motion between the front and rear parts. Empty, the vehicle weighs about 13 tons, and when fully laden, it weighs about 25 tons, but dual tyres are used only for the driving axle, which carries the heaviest load.

Power unit

Motive power is supplied by a Saurer type CV1D, twelve-cylinder V-type engine, developing 240 b.h.p. at 2,000 r.p.m. The cylinder bore and stroke are 115 and 140 mm respectively and the swept volume is 17·46 litres. The turbo-charged version of this engine was described in detail in the January 1948 issue of *Automobile Engineer*. This was the CVD unit, with the same stroke but a bore of 110 mm and developing 285 b.h.p. when turbo-charged, as compared with 220 b.h.p. at 2,000 r.p.m. in the normally aspirated condition. The cylinders are arranged in two banks at 60 deg. Wet-type, cast iron cylinder liners are spigoted in the crankcase and clamped by the cylinder head. The liners and the connecting rod assemblies can be withdrawn after the head has been lifted and big end bearing caps removed.

The crankshaft is a fabricated component, built up of six separate throws, bolted together at their abutting circular webs. It is carried in seven roller bearings, two being installed side by side at the centre. The valves are actuated by rockers, push rods and a single camshaft accommodated in a tunnel in the vee of the cylinder banks. Dry sump lubrication is employed, two pumps scavenge the sump and circulate the oil through a water-cooled heat exchanger, whence it passes to finned longitudinal containers at each side of the crankcase.

With its cast iron block, the engine weighs about 2,730 lb. The maximum torque of 695 lb-ft is developed at 1,400 r.p.m. Naturally, the well-known toroidal combustion chamber originated by the late H. Saurer is used. When the engine is run on gas oil with a calorific value of 19,000 B.Th.U/lb, the minimum fuel consumption is about 0·375-0·39 lb/b.h.p.-hr.

The engine, together with its

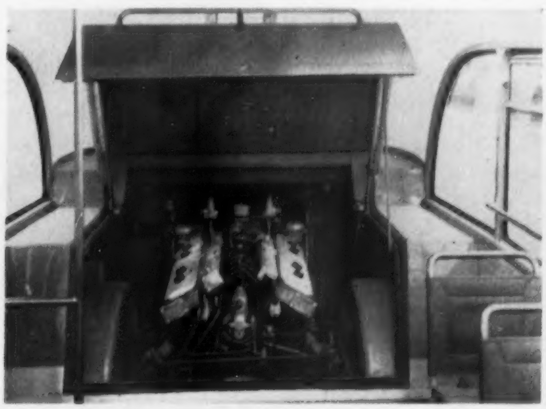


Fig. 5. Spring assistance is provided for lifting the engine cover to give access to the power unit from inside the vehicle

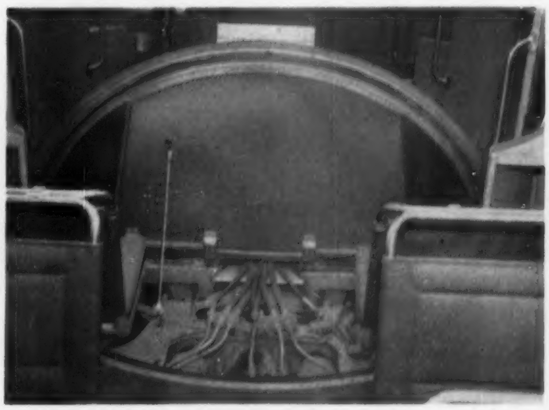


Fig. 6. The floor plates over the turntable are hinged to give access to the pipe connections beneath them

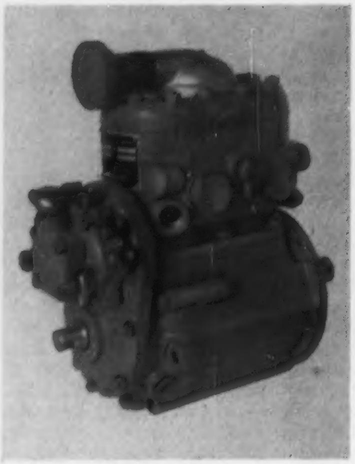


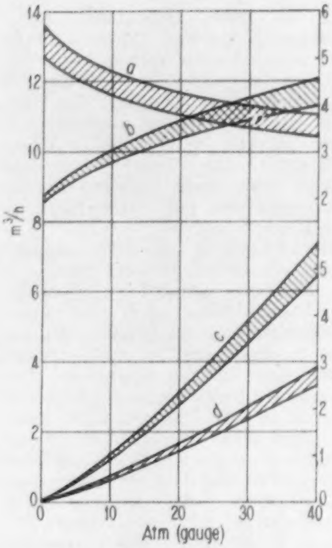
Fig. 7. The two-cylinder compressor

radiator and a single cast aluminium fan, which is belt driven from the crankshaft, is installed at the rear of the vehicle. Cooling air is taken in at the rear through venturi-shaped passages between louvres, and discharged downwards after passing through the radiator and over the engine. The coolant temperature is thermostatically controlled and, in addition, the radiator is provided with thermostatically controlled, pneumatically operated shutters. Access to the engine compartment is gained from the outside of the vehicle through hinged doors that extend round the rear quarters, Fig. 4. From the inside of the vehicle, access can be gained to the engine by lifting a hinged cover, which is spring-balanced to ensure easy handling, Fig. 5. To reduce transmission of engine noise into the body, the cover is lined with sound absorbing material. It is also rigid enough to carry luggage as well as passengers sitting on it occasionally during rush hours. Two lamps are provided in the engine compartment.

Compressed air system

In common with the practice adopted with most Swiss buses, the engine is started pneumatically. This arrangement is favoured in Zurich because it reduces overall vehicle weight since only a small battery is needed; at the same time troubles due to starter pinion wear are eliminated. Other municipal undertakings, such as that at Berne, stipulate that the engine must be cut at every stop, even in city service, in order to conserve fuel; this would call for unduly large batteries, if electric starting were used.

A high-pressure air system is employed, Fig. 3. It is served by a two-cylinder compressor, Fig. 7, each cylinder giving two-stage compression to a final pressure of 590 lb/in^2. The entire installation is supplied by the Nova-Werke (Junker and Ferber) of Zurich. It is of interest to compare the relative merits of the high and low pressure systems. Conventional low pressure air systems usually operate at a pressure of 90 lb/in^2, the compressor being unloaded at about 120 lb/in^2. However, with the high-pressure system used on this vehicle, a pressure reducing valve maintains a practically constant pressure in the brake system and thus ensures satisfactory operation under all conditions. In the new bus,



a Delivery volume
b Power consumption
c Charging time, 30 litres
d Charging time, 15 litres

Fig. 8. Performance curves for air compressor running at 1,000 r.p.m. Its specification is: two cylinder, bore 68 mm, stroke 42 mm, swept volume 152·5 cm^3

air at 590 lb/in^2 is stored in two 40 litre, light alloy cylinders mounted on the right at the front end of the vehicle, where they are easily accessible through a hinged exterior panel, Fig. 10. Since brakes tend to become unreliable if the pressure falls below about 65 lb/in^2, the effective air capacity of this system amounts to $2\times40\times(590-65)=42{,}000$, as compared

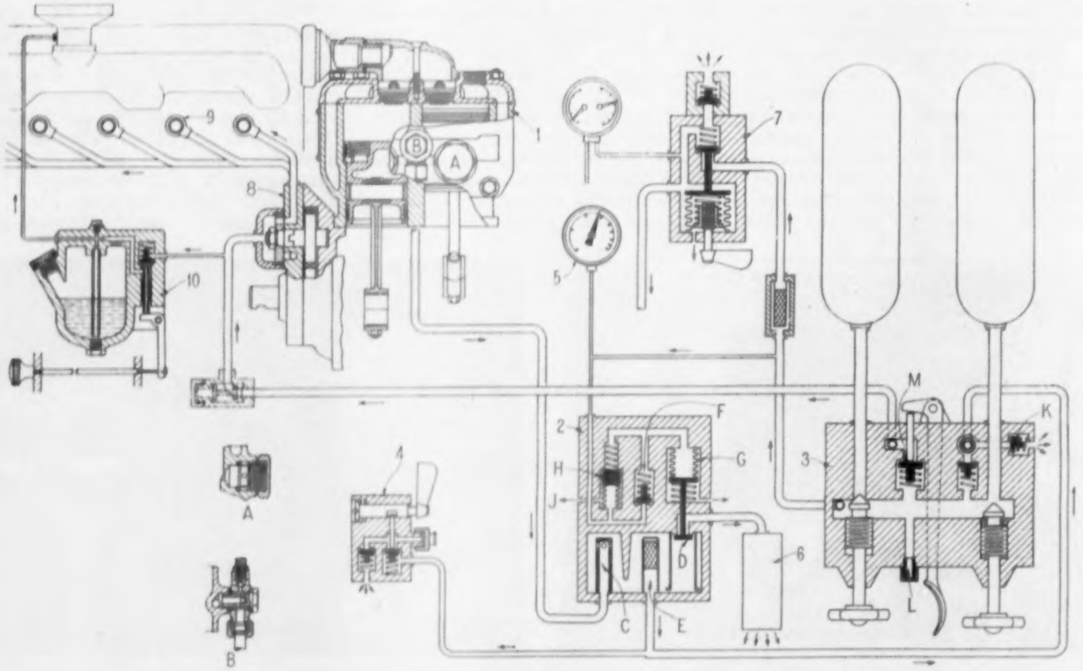


Fig. 9. Diagramatic illustration of the layout of some of the main components of the pneumatic system

with only $2 \times 40 \times (120-65) = 4{,}400$ in an equivalent low-pressure system.

A further advantage is the reduced air consumption obtained. This reduction is effected because with most air-operated units it is necessary to charge both the operating cylinder and the feed line; if the relevant volume is, say, 0·1 litres, the air consumption per operation is 0·1 litres × pressure. With most low-pressure systems the pressure varies between 90 and 120 lb/in², the average value being about 105 lb/in². On the other hand, with the high pressure arrangement, a suitable pressure reducing valve maintains a pressure of about 85 lb/in² in the brake circuit, and 62·5 lb/in² in the door-motor and windscreen-wiper systems. Thus, the air consumptions are, respectively, 18 and 39·3 per cent smaller than with a low-pressure air system, the average air consumption being reduced by about 20 per cent. Against this, it must be admitted that the power required to drive the high-pressure compressor is some 40 per cent higher than that for a low-pressure unit, but since the amount of air required is some 20 per cent less, the extra fuel consumed is only 12 per cent.

The high-pressure installation is shown diagrammatically in Fig. 9. A two-stage compressor draws its air supply through the engine air cleaner. After being compressed at a ratio of about 7:1 in the first stage, the air is forced into the annular space in which the second stage of compression is effected. In some designs the transfer between the first and second stages is

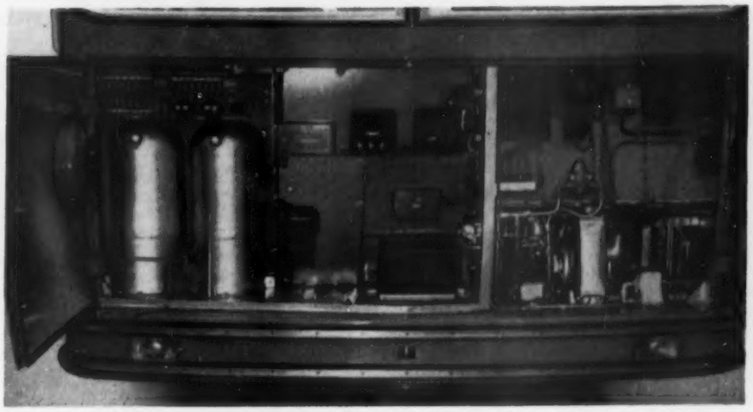


Fig. 10. Accessibility of the electric and pneumatic units at the front end is a noteworthy feature

effected past two non-return valves in a passage in the cylinder block and head, but in the two-cylinder compressor used with the new bus, it is effected through a non-return valve in the piston. The air leaves the compressor, through the non-return valve at A, into the pipe connected at B. This pipe takes it to the high-pressure regulator 2, which it enters through a nozzle C in the oil separator space. If the valve D is closed, the air leaves through the filter E to the control unit 3 and the tyre valve 4. Pressure from the line between the control unit and the gauge 5 is led to the valve F, which regulates the maximum pressure. As soon as the pressure attains the value of 560 to 590 lb/in², this valve lifts off its seat, and the pressure reaches the bellows G and opens valve D, thus unloading the compressor and discharging air and condensate from the chamber to atmosphere through the silencer 6.

When the pressure drops to 440 to 470 lb/in², valve H is forced down by its spring. This releases air from bellows G to the atmosphere at J, and valve D closes. In this condition, the compressor charges the bottles by way of the control unit 3. This control unit incorporates manually-operated bottle cut-off valves; it is recommended that one bottle is used for normal service, while the other is kept fully charged as a stand-by. A safety valve K, set to blow off at a pressure some 70 lb/in²

Fig. 11. In the four-speed gearbox, oil-actuated multi-plate clutches are incorporated for the selection of the gears

above maximum pressure, is incorporated, in addition to the non-return valve in the control unit. Provision is made, at L, for connection to an outside source of supply, in case the pressure in the bottles becomes unduly low.

The setting of the pressure reducing valve 7 for the brakes and auxiliaries can be adjusted by means of a small hand wheel on the unit, which is also provided with a safety valve. Air for starting passes through a valve M in the control unit 3 and a rotary distributor 8, to the non-return valves 9 in the engine cylinders. If, as in some other designs, the compressor is gear-driven, the distributor valve is incorporated in it; both the compressor and distributor in this case run at half engine speed. When the engine is running, the rotor floats about 0·5 mm away from the housing face. This ensures the maintenance of an effective oil film between the rotor and its casing. However, when the engine is stationary and air is admitted to start it, the valve is forced against the distributor face.

If a belt-driven compressor is used, as in the Zurich bus, the distributor valve must be positively driven by the engine. In this vehicle, only ten of the twelve cylinders are equipped for air starting, and the distributor valve is made up of two five-line units in tandem on a common shaft. To facilitate starting, a vaporizer 10, filled with six parts of gas oil and one of ether, can also be provided, but although this is of some advantage for long-distance lorries, it is not normally required for city buses. The output characteristics and power requirements of the two-cylinder, 305 cm^3 compressor running at the top speed of 1,000 r.p.m. are shown in Fig. 8.

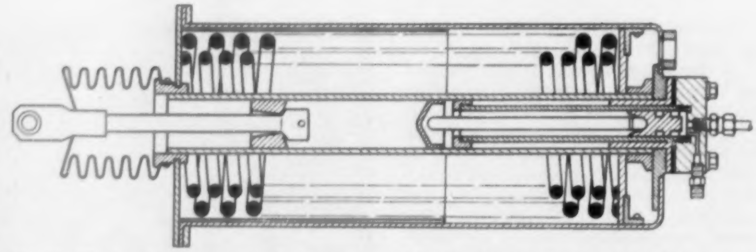


Fig. 12. The hand brakes are applied by concentric coil springs, and released by air pressure. In the event of the air pressure being too low, the brakes can be released by pumping oil into a small hydraulic cylinder mounted in the end of the pneumatic unit

Transmission

A vibration damping coupling with a non-linear characteristic is incorporated in the transmission. This non-linearity is obtained by an ingenious combination of leaf springs and rollers, the effective radius of load application in the coupling varying with the torque. From this coupling the drive is taken by a cardan shaft to a four-speed gearbox of novel design, Fig. 11. In the box, there is an oil-operated multi-plate clutch for each gear. The gears are electrically pre-selected by a lever on the left of the driver and then engaged by depressing the clutch pedal. To ensure silent running, helical gears are used. All the shafts run in ball bearings. Pressure lubrication with oil delivery through nozzles is employed. The overall transmission efficiency varies between 94 and 96 per cent, according to which gear is engaged.

The main advantages of this gearbox are: its suitability for remote control; the clutches require neither setting nor adjustment; rapid gear changes; simple and robust construction with resultant simplicity of operation and maintenance; and two clutches share the slip action during take-up from rest and, consequently, clutch wear is low. The gear ratios are 3·78:1, 2·44:1, 1·51:1 and 0·98:1, the corresponding maximum road speeds being 8·75, 14·35, 22·5 and 34·4 m.p.h.; in reverse, the speed is limited to 8·75 m.p.h.

This vehicle is unusual in that the engine and gearbox are in the trailer portion but they drive the rear axle of the front portion. Therefore, the cardan shaft drive from the gearbox to the double reduction axle has to pass forward under the articulated joint, Fig. 16. The first reduction is by a conventional spiral bevel gear and the second is by spur gears adjacent to the wheels at each side, Fig. 13. This arrangement gives a low floor level since the spiral bevel unit and half shafts are lower than the wheel hubs. The overall axle ratio is 7·8:1.

Chassis

The frames of both portions of the vehicle are of welded construction throughout. A special universal coupling is employed between them. The tractive effort is transmitted through two bushes between each half of the vehicle and the turntable; these allow an angular displacement of up to 9 deg in elevation. To help to support the concertina type cover over the passage between the front and rear portions of the vehicle, a portal frame is mounted on a cross member secured to the turntable. This cross member is linked to the two chassis frames by a connecting rod and a knee-shaped lever. It also carries the turntable cover, which is diametrically divided, the two parts being hinged together to facilitate inspection of the air, oil and electric lines led through beneath them, as well as to accommodate relative motion between the front and rear portions

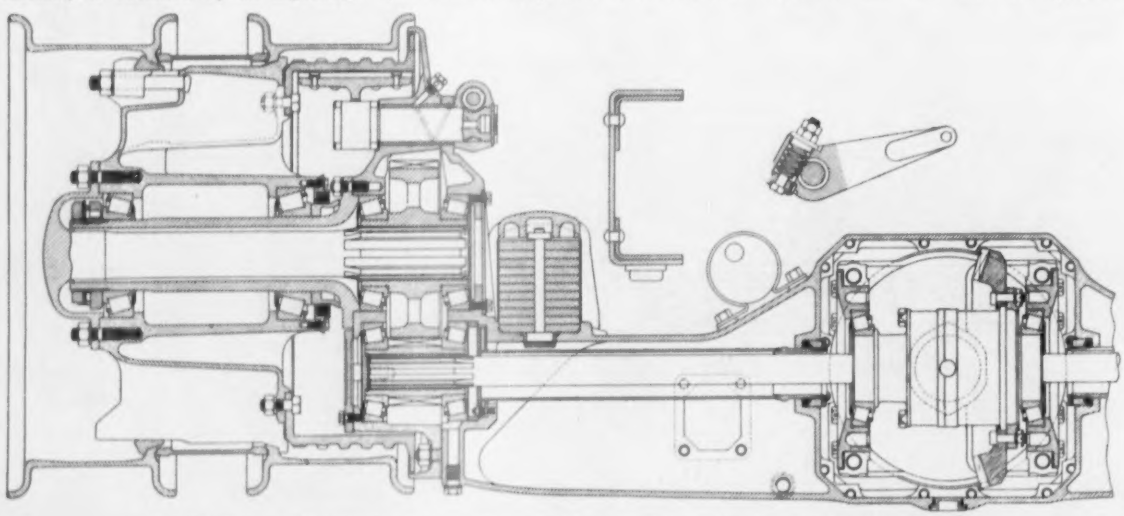


Fig. 13. The double-reduction rear axle is cranked downwards at the centre to reduce floor height

Fig. 14. The control for the exhaust brake is on the dash, to the right of the steering column, and those for the pneumatic starter and the hydraulic pump for releasing the hand brake are on the left of the driver

of the vehicle as it runs over road irregularities and negotiates changing gradients. Particular care has been taken to prevent the entry of draught, dust or water by providing rubber aprons round the outer edges of the turntable cover, and dense brush seals on the underside of it, Fig. 6.

For the suspension, semi-elliptic springs are used throughout. The springs of the driving axle have a non-linear characteristic, which is obtained by the use of curved slider pads at one end. Since the rear portion is a single-axle trailer, there is a possibility of jack-knifing, particularly when the vehicle is being reversed. To minimize the danger in the event of jack-knifing occurring, a contact that operates a buzzer in the driver's compartment is energized should the angle between the two portions be reduced below a certain value. This warns the driver to take corrective action. The minimum turning radius, as measured to the outer of the two front wheels, is 32 ft 9 in, and as measured to the outer corner at the front of the body of the forward portion, is 37 ft 6 in.

On all six wheels, the brakes are air-operated, but to prevent the rear part from overriding the front, the brakes of the rear wheels are applied a fraction of a second before the others. Vehicle speed on downgrades, or when slowing down gently prior to coming to a stop, is reduced by application of the exhaust brake, which is pneumatically operated. This brake is controlled by a lever and an electro-pneumatic valve under the right-hand side of the steering wheel, Fig. 14.

The hand brake is applied on the driving wheels only. Its operating mechanism comprises two cylinders containing helical springs which, to release the brakes, are compressed by an air-operated piston. For parking, the brakes are applied by releasing the air. If, while the vehicle is parked, the air pressure should fall to such level that it is impossible to release the brakes again pneumatically, the springs can be compressed by applying oil pressure to small auxiliary pistons incorporated in the main cylinder units, Fig. 12. The pressure is supplied by a manually operated pump in the driver's cab. When applying the parking brake, the air brakes on the rear wheels are also applied automatically, again slightly before the spring-actuated units. The air for the brakes, door control, etc., is taken from the high pressure system, as shown in Fig. 3. To ensure effective starting and braking under all conditions, electro-pneumatically operated sanders are provided, by means of which sand is blown on to the ground in front of the driving wheels. Three auxiliary low pressure reservoirs are provided in addition to the two high pressure bottles.

Bodywork

Most of the body frame is of aluminium sections bolted to the chassis frame. It is covered with light alloy panels so secured as to permit easy replacement in case of damage. At the front and rear, steel framing and panels are employed. The interior finish is in light grey up to the cant rail and the ceiling panel is white. Light brown trim is used on the seats.

Fig. 15. The conductor's seat is adjacent to the turntable

Interior heating is effected by the engine coolant. Two Olta air conditioning units, one for the front and the other for the rear portion of the vehicle, supply hot air through a duct on each side just above floor level. In the sides of these ducts there are a number of openings through which the warm air is discharged into the body. Separate heaters are provided in the driver's compartment and under the conductor's seat. They have variable speed blowers and are controlled directly by the driver and conductor.

Fresh air for ventilation only is taken in through an opening over the front destination signs. It is directed into a duct immediately under the roof of the front portion of the vehicle and

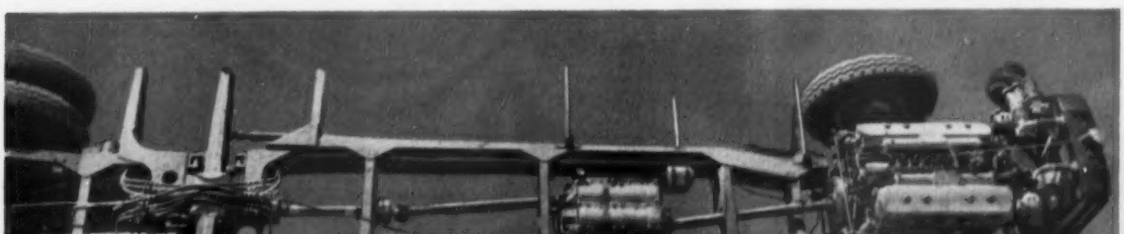


Fig. 16. The engine is installed at the back of the trailer and drives the rear axle of the front portion of the vehicle

Fig. 17. A view from the rear portion of the vehicle, looking forward to the driver's compartment. Tinted glass is employed in the partition behind the driver

Fig. 18. The rear portion is shorter than the front one and a greater proportion of the space in it is for standing passengers so that loading can be effected rapidly

is discharged to the interior through three openings. These openings can also be connected directly to louvres immediately above them on the roof by means of a three-position, hinged flap valve. The three positions are: air from front intake, air from roof aperture, and shut. Two similar ventilators are provided in the rear unit. Additional ventilation is through hinged panels on the windows. These can be shut by the passengers but opened only by the conductor with a special key that also fits the roof ventilator flap control.

A signal pillar divides the windscreen vertically at the centre. It carries light signals that indicate: dynamo charging, coolant temperature, air pressure in the tyres while they are being inflated, oil pressure and start and stop signals. The signals are given by a green light that is switched on by passengers or the conductor to indicate to the driver that he is required to stop; the starting signal is given by the conductor switching this light off. To show to the conductor and the passengers that a stop signal has been given, other green lights are illuminated simultaneously, one at the conductor's desk, two in the front portion of the vehicle and one in the rear.

Fig. 19. The seats are higher than usual and are equipped with footrests

All doors are air-operated. The rear and centre doors are controlled by the conductor, while the front door is operated by the driver. An air-operated sliding door is installed between the driver's platform and the rest of the vehicle, Fig. 17. It is to isolate the driver from passengers during rush-hour traffic. This door, together with the two front partition windows, is fitted with dark blue glass, to make the driver's task easier at night.

A total of 43 seats can be installed, but to provide more standing room this number has been reduced to 30, Figs. 2, 17 and 18. The seats are higher than usual and are equipped with foot rests, Fig. 19. This arrangement has been adopted primarily for elderly passengers: leaving these seats is easy because the passengers do not have to lift themselves as they do when leaving the lower, more conventional types of seat. A single bolt, which can be reached with a box spanner after removing the seat cushion, is employed to anchor the seat to the floor.

The conductor sits near the rear entrance door, Fig. 15. He has a small table for tickets and a switchboard on which are the door control and signal switches. So that he has a clear view of the centre door, two seats are installed on the turntable section adjacent to the conductor's station; this prevents obstruction of the conductor's line of vision by standing passengers. The vehicle is equipped with six loudspeakers mounted under the roof. They are used for announcements by the driver or the conductor, who are each provided with a microphone; the loudspeakers are switched in by a foot switch. Two more loudspeakers are mounted outside at the side of the front destination sign so that the crew can inform the passengers waiting at stops as to the route and destination of the vehicle. These are not used after eight o'clock in the evening lest they disturb local residents.

Electrical equipment

The electrical circuit is of considerable interest. Because the relatively high power demand of 2 kW could not be met by available D.C. generators, an A.C. unit is employed. It is belt-driven from the transmission shaft. The generator charges two 150 amp-hr batteries; a Selenium rectifier and regulator is incorporated in the circuit. Interior lighting is by ten fluorescent lamps in a 220 volt, A.C. circuit, supplied by two motor generators, each feeding five lights. Seven of the lighting units are in the front portion and the remainder are at the rear. The flashing direction indicators are arranged in pairs one above the other in a common housing. There are three pairs on each side, one at each end and one in the middle. To ensure reliable operation they are controlled by a 90 W motor-driven switch. The upper and lower bulbs of each unit are switched on and off alternately. This arrangement has been found to be most effective.

RECORD CAMERA

THE Fairchild Oscillo-record camera is now available through the sole British agents, Polarizers (U.K.) Ltd., 186 Acton Lane, London, N.W.10, for recording cathode ray oscilloscope images. This camera, the first to be designed specifically for the purpose, extends the usefulness of the oscilloscope. It produces photographic records of high-speed transients, stationary patterns of periodically returning phenomena, or any type of phenomena that can be put on a cathode ray oscilloscope. It affords a satisfactory method of studying non-recurring phenomena which occur either too rapidly to permit adequate study, or too slowly to give continuity. Combinations of very slow-speed phenomena with occasional high-speed transients can also be readily studied.

STROKE:BORE RATIO

The Influence of Stroke:Bore Ratio on Crank Pin Bearing Loads

Edward G. Ingram

SOME years ago the author investigated the subject of stroke : bore ratio, with particular attention to relative crank pin bearing loads in long- and short-stroke engines with equal piston displacements.* The analysis, which was carried out with the aid of polar force diagrams, indicated that at any specified r.p.m. the mean bearing load resulting from the combined effect of inertia, gas pressure, and centrifugal forces throughout the cycle was lower in the short-stroke engine at high speeds, because the shorter crank radius results in a lower centrifugal load.

Certain assumptions were made in this theoretical study, one being that the inertia force in the long- and short-stroke forms would be the same for corresponding crank positions because the larger diameter, heavier piston of the short-stroke engine would move at a lower piston speed. Because few short-stroke engines were in existence at that time, satisfactory information as to how the weights of short-stroke engine pistons compare with those of long-stroke engines in actual practice was not then available.

It is believed that a reconsideration of the whole subject in the light of modern design trends will be of interest, especially in view of the recent movement towards the use of very short-stroke engines.

The polar diagrams presented in Figs. 1 and 2 show the forces acting on the crank pin bearings of two engines with cylinders of approximately the same piston displacement, but with stroke-bore ratios of 1·58 to 1 and 0·88 to 1, respectively. The operating speed is 4,000 r.p.m. These engines, which are designated respectively as Engine No. 1 and Engine No. 14, while in a sense hypothetical, are based on the dimensions of actual automobile engines of present-day design with respect to cylinder bore and stroke, weight of reciprocating parts, and ratio of connecting rod length to crank radius—Table VI gives the dimensions of other modern or fairly recently designed engines dealt with in the same way. Since only one cylinder and the load on its crank pin bearing are being considered, it makes no difference whether one engine is a six, for example, and the other a V-eight, as is the case with Engines No. 1 and No. 14. The force diagrams are shown for single vertical cylinders in both cases.

For the compression and expansion strokes, the gas pressures are assumed to be the same for all engines discussed in this article. They are for a cylinder with 7·5 to 1 compression ratio, and are given at 30-degree crank intervals for full throttle operation at peak speed. These pressures were not obtained from an indicator diagram, but were supplied to the writer through the courtesy of P. M. Heldt.

The figures were calculated from the b.m.e.p. that is estimated to be typical for engines with a 7·5 to 1 compression ratio. On the expansion stroke, a peak pressure of approximately 600 lb/in² is reached at about 18 deg past top dead centre. Because the influence of this peak pressure on the crank pin bearing load is important to the investigation, it has been necessary to include an extra crank position, point 13A, in the force diagrams.

The crank pin is affected by the action and reaction of three forces: the gas pressure, E, acting on the piston head; the inertia force, I, due to the reciprocating masses, which include the piston and upper part of the connecting rod; and the centrifugal force, C, due to the lower part of the connecting rod and its bushing, which may be considered as rotating weight.

In comparing long- and short-stroke cylinders of the same piston displacement, force E at any point in the stroke will always be greater in the short-stroke cylinder, the amount being directly proportional to the area of the piston.

The formula for determining the inertia force is

$$I = 0{\cdot}0000142WSN^2 \left(\cos A + \frac{r}{L}\cos 2A\right)$$

where W is the weight of the reciprocating parts in pounds; S is the length of the stroke in inches; N is the number of revolutions per minute; A is the crank angle in degrees; r is the crank radius in inches; and L is the length of the connecting rod in inches. The expression in the parentheses is termed the inertia factor, and its value for

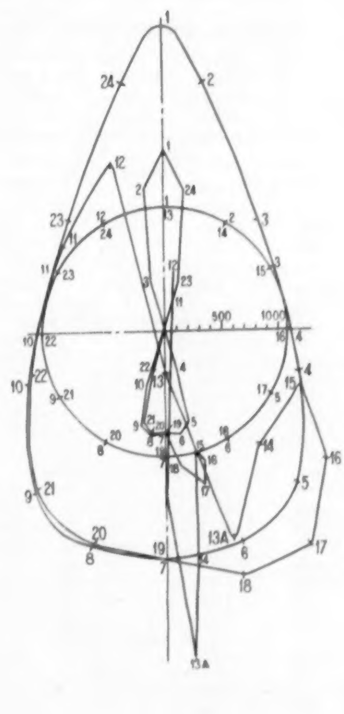


Fig. 1, left. Polar diagram of forces acting on the crank pin bearing of a long-stroke engine, stroke:bore ratio 1·58:1

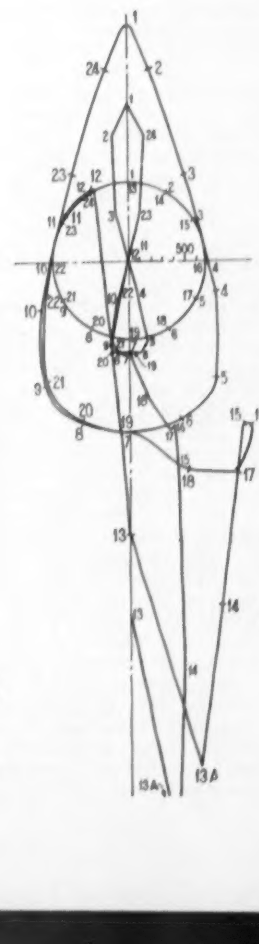


Fig. 2, right. Forces acting on the crank pin bearing of a short-stroke engine, stroke: bore ratio 0·88:1

* "*Analyzing the Question of Stroke-Bore Ratio,*" *Automotive Industries, Nov. 20, 1919.*

TABLE I. LONG-STROKE ENGINE

No.	Crank position, deg.	E	I	E+I	H	C_v	C_h	(E+I) $+C_v$	H+C_h	T
Inlet Stroke										
1	0	0	+1567	+1567	0	+1077	0	+2644	0	2644
2	30	0	+1230	+1230	−177	+933	+538	+2163	+361	2193
3	60	0	+435	+435	−111	+538	+933	+973	+822	1277
4	90	0	−349	−349	+104	0	+1077	−349	+1181	1230
5	120	0	−784	−784	+200	−538	+933	−1322	+1133	1755
6	150	0	−882	−882	+127	−933	+538	−1815	+665	1925
Compression Stroke										
7	180	0	−871	−871	0	−1077	0	−1948	0	1948
8	210	−7	−882	−889	−128	−933	−538	−1822	−666	1950
9	240	−35	−784	−819	−209	−538	−933	−1357	−1142	1775
10	270	−85	−349	−434	−129	0	−1077	−434	−1206	1275
11	300	−226	+435	+209	+53	+538	−933	−747	−880	1155
12	330	−707	+1230	+523	+76	+933	−538	+1456	−462	1525
Expansion Stroke										
13	360	−3004	+1567	−1437	0	+1077	0	−360	0	360
13A	378	−4241	+1441	−2800	+248	+1024	+333	−1776	+581	1869
14	390	−3146	+1230	−1916	+277	+933	+538	−983	+815	1275
15	420	−1449	+435	−1014	+259	+538	+933	−476	+1192	1280
16	450	−763	−349	−1112	+333	0	+1077	−1112	−1410	1800
17	480	−530	−784	−1310	+334	−538	+933	−1848	+1267	2245
18	510	−282	−882	−1164	+127	−938	+538	−2097	+665	2200
Exhaust Stroke										
19	540	0	−871	−871	0	−1077	0	−1948	0	1948
20	570	0	−882	−882	−127	−933	−538	−1815	−655	1925
21	600	0	−784	−784	−200	−538	−933	−1322	−1133	1755
22	630	0	−349	−349	−104	0	−1077	−349	−1181	1230
23	660	0	+435	+435	+111	+538	−933	+973	−822	1277
24	690	0	+1230	+1230	+177	+933	−538	+2163	−361	2193

TABLE II. SHORT-STROKE ENGINE

No.	Crank position, deg.	E	I	E+I	H	C_v	C_h	(E+I) $+C_v$	H+C_h	T
Inlet Stroke										
1	0	0	+1363	+1363	0	+671	0	+2034	0	2034
2	30	0	+1074	+1074	−144	+581	+335	+1655	+191	1662
3	60	0	+395	+395	−94	+335	+581	+730	+487	890
4	90	0	−286	−286	+79	0	+671	−286	+750	802
5	120	0	−681	−681	+161	−335	+581	−1016	+742	1260
6	150	0	−789	−789	+106	−581	+335	−1370	+441	1440
Compression Stroke										
7	180	0	−790	−790	0	−671	0	−1461	0	1461
8	210	−10	−789	−799	−108	−581	−335	−1381	−443	1450
9	240	−51	−681	−732	−174	−335	−581	−1067	−775	1300
10	270	−124	−286	−410	−113	0	−671	−410	−784	887
11	300	−130	+395	+65	+15	+335	−581	+400	−566	695
12	330	−1032	+1074	+42	+6	+581	−335	+623	−329	700
Expansion Stroke										
13	360	−4386	+1363	−3023	0	+671	0	−2351	0	2351
13A	378	−6192	+1254	−4938	+409	+637	+207	−4301	+616	4345
14	390	−4592	+1074	−3522	+474	+581	+335	−2941	+809	3075
15	420	−2116	+395	−1721	+409	+335	+581	−1386	+990	1712
16	450	−1114	−286	−1400	+387	0	+671	−1400	+1058	1765
17	480	−774	−681	−1455	+345	−335	+581	−1790	+926	2025
18	510	−413	−789	−1194	+161	−581	+335	−1775	+496	2250
Exhaust Stroke										
19	540	0	−790	−790	0	−671	0	−461	0	1461
20	570	0	−789	−789	−106	−581	−335	−1370	−441	1440
21	600	0	−681	−681	−161	−335	−581	−1016	−742	1260
22	630	0	−286	−286	−79	0	−671	−286	−750	802
23	660	0	+395	+395	+94	+335	−581	+730	−487	890
24	690	0	+1074	+1074	+144	+581	−335	+1655	−191	1662

E, gas pressure.
I, Inertia force.
H, horizontal component or (E+I) tan connecting rod angle to vertical.
C_v, vertical component of centrifugal force.
C_h, horizontal component of centrifugal force.
(E+I)+C_v, total for vertical components.
H+C_h, total for horizontal components.
T, resultant load on crank pin (approximate).

different values of $\frac{r}{L}$ and crank angle may be obtained from published tables.

In the engines discussed, the reciprocating weight is taken to be the weight of the piston plus one-third the weight connecting rod, rather than the weight of piston plus the weight of the upper one-third of the rod length. Although the rod is, of course, somewhat larger and heavier at the lower end, this arbitrary division of weight is considered a fair approximation for two reasons: first, because the weight of the piston rings is not included, and secondly, because some designers consider the upper one-half of the rod length as reciprocating weight.

The formula for determining the centrifugal force is

$$C = 0{\cdot}00034wRN^2$$

where w is the weight of the rotating mass in pounds; R is the crank radius in feet; and N is the revolutions per minute. The rotating mass is here assumed to be two-thirds the weight of the connecting rod. In this analysis, vertical force components are positive when acting upward, negative when acting downward. Horizontal force components are positive when acting to right, negative when acting to left. As viewed, the engines turn clockwise.

The direction of the centrifugal force is in phase with the crank pin, and therefore represented in the diagram by a concentric circle for one revolution. Two congruent circles constitute a cycle. The zigzag curve represents the force resulting from gas pressure, the inertia of the reciprocating parts, and the angularity of the connecting rod. It can be seen that the approximate total load on the crank pin and its bearing for the different crank positions—the outer of the curves—is found by combining graphically the forces of the zigzag curve with those represented by the circle. Details of the calculations are given in Tables I and II.

Examination of the resultant curves reveals that, except during the expansion stroke, crankpin bearing loads are lower in the short-stroke engine. In fact, the mean bearing load for 25 crank positions tabulated is approximately 5·7 per cent lower in the short-stroke form. If point 13A had been omitted in both engines, the mean bearing load would have been 12·1 per cent lower.

A knowledge of the maximum force that occurs during the cycle is important when determining stresses and deciding on bearing dimensions. Early automobile engine designers used the maximum expansion force for this purpose. But it was soon found that with the long-stroke engines then in general use the force resulting from the inertia of the reciprocating parts combined with centrifugal action at top dead centre of the induction stroke was greater.

The recent introduction of very short-stroke engines has changed the situation. Table II and the polar diagram for short-stroke engine No. 14 show that the total crank pin bearing load at top dead centre of the induction

TABLE IV.

No. 1X	1·35	2439	2440
No. 14X	1·40	2609	2551
No. 15X	1·11	2696	2730

stroke is 2,034 lb, while at 18 deg past top centre of the expansion stroke it is 4,345 lb, or more than twice as great. This is the reverse of the situation with the long-stroke engine No. 1. Table I shows that the crank pin bearing load at top dead centre of the induction stroke is 2,644 lb, as against 1,869 lb at 18 deg past top dead centre of the expansion stroke. Because the foregoing figures only cover two engines, more convincing evidence is found in Table III, which gives the crank pin bearing loads at top dead centre of the induction stroke and at 18 deg past top dead centre of the expansion stroke for twelve engines, of recent or fairly recent design, with stroke:bore ratios ranging from 1·58:1 to 0·80:1, computed on the basis of the same gas pressures and an operating speed of 4,000 r.p.m.

It can be seen that in the engines with stroke:bore ratios between 1·58:1 and 1·25:1 the crank pin bearing load at top dead centre of the induction stroke is higher than at 18 deg past top dead centre of the expansion stroke, but that the reverse is true for engines with stroke:bore ratios from 0·95:1 to

TABLE III.

Engine	Stroke : bore ratio	Crank pin bearing load at T.D.C. inlet stroke	Crank pin bearing load at 18 deg past T.D.C. expansion stroke
No. 1	1·58	2644	1869
No. 2	1·42	3352	1970
No. 3	1·38	3896	2068
No. 4	1·35	3379	1940
No. 6	1·25	3488	2422
No. 7	1·25	3409	2684
No. 9	0·99	2922	3469
No. 10	0·92	2527	3890
No. 11	0·92	2776	4200
No. 14	0·88	2034	4345
No. 15	0·86	2401	3980
No. 16	0·80	2145	5591

0·80:1. The crank pin bearing load for the engine with the lowest stroke: bore ratio is more than 2½ times as great as the load at top dead centre of inlet stroke.

In general, lowering the stroke:bore ratio of a long-stroke engine, within certain limits, tends to decrease the relatively high crank pin bearing load at the top dead centre of the inlet stroke and increase the relatively low load at 18 deg past top dead centre of the expansion stroke. Conversely, increasing the stroke:bore ratio of a short-stroke engine, within certain limits, tends to increase the relatively low bearing load at top dead centre of the inlet stroke and decrease the relatively high load at 18 deg past top dead centre of the expansion stroke. This being the case, it should be possible to find a stroke: bore ratio for a particular engine which would make the loads for both points about equal, and thus reduce the maximum load occurring during the cycle. But to do this it is necessary to know how the weight of the reciprocating parts of a particular engine vary with the stroke:bore ratio.

In the author's original article, in which an actual long-stroke engine was compared with a hypothetical short-stroke unit of the same piston displacement, it was assumed, as already mentioned, that because the larger diameter, heavier piston of the short-stroke engine moved at a lower piston speed for a given r.p.m., the inertia force at corresponding crank positions would be the same as in the long stroke engine. This matter will now be considered further.

With cylinders of the same piston displacement and with the ratio of connecting rod length:crank radius the same, if the inertia force is to be equal to that of a given long-stroke engine at equal r.p.m., and corresponding crank angles, the reciprocating weight, W_t, of a short-stroke engine is given by $W_t = \frac{S_l}{S_s} W_l$

In this equation S_l is the stroke of the long-stroke engine, S_s, the stroke of the short-stroke engine, and W_l is the reciprocating weight of the long-stroke engine.

The equation

$$d = \frac{W_s - W_t}{W_t} \times 100$$

gives the deviation, d, in per cent, of the actual reciprocating weight of the short-stroke engine, W_s, from the theoretical weight, W_t, required to make the inertia force at corresponding crank positions the same as in the long-stroke engine. Positive deviations mean the inertia force in the short-stroke engine is higher than in the long-stroke engine, negative deviations lower.

Table V contains reciprocating weight comparisons, based on the above formulæ, for automobile engines of recent or fairly recent design. It covers pairs of engines with cylinders having different stroke: bore ratios but about the same piston displacement for each particular pair.

The assumption that the rod:crank-radius ratio is the same is not quite true for the engines listed, for the ratio varies from about 3½:1 to 4:1, but the error is not large enough seriously

TABLE V.

Engine	Stroke: bore ratio	Piston displacement of one cylinder	W_l	W_t	W_s	$\frac{W_s - W_t}{W_t} \times 100$
		Comparison No. 1				
No. 8	1·18	29·9	19·38			
No. 13	0·89	29·9		23·45	25·80	+10·0
		Comparison No. 2				
No. 5	1·25	31·9	19·28			
No. 15	0·86	32·0		24·87	30·92	+24·3
		Comparison No. 3				
No. 1	1·58	33·6	18·10			
No. 14	0·88	33·0		26·86	23·60	−12·1
		Comparison No. 4				
No. 1	1·58	33·6	18·10			
No. 15	0·86	32·0		27·69	30·92	+11·7
		Comparison No. 5				
No. 4	1·35	36·3	26·37			
No. 8	0·92	34·5		34·54	25·90	−25·0
		Comparison No. 6				
No. 2	1·42	38·3	25·30			
No. 11	0·92	39·6		33·40	31·85	−4·7
		Comparison No. 7				
No. 2	1·42	38·4	25·30			
No. 9	0·99	37·1		32·45	29·10	−10·3
		Comparison No. 8				
No. 6	1·25	40·0	30·00			
No. 16	0·80	40·2		40·38	29·19	−28·0
		Comparison No. 9				
No. 6	1·25	40·0	30·00			
No. 12	0·90	40·5		37·62	31·55	−16·1
		Comparison No. 10				
No. 7	1·25	42·0	29·01			
No. 11	0·92	39·6		36·26	31·85	−12·2

W_l is the reciprocating weight of long-stroke engine, oz. W_t is the reciprocating weight of short-stroke engine required to give inertia force equal to that of long-stroke engine, oz. W_s is the actual reciprocating weight of short-stroke engine, oz. $\frac{W_s - W_t}{W_t} \times 100$ is the deviation from the reciprocating weight for equal inertia force, per cent.

TABLE VI.

Engine	Stroke : bore ratio	Piston displacement of one Cylinder	Bore and stroke, in	Piston weight, lb	Connecting rod weight, oz	Connecting rod length, in.
No. 1	1·58	33·6	3·00 × 4·75	10·13	24·05	$8\frac{3}{16}$
No. 2	1·42	38·4	3·25 × 4·63	16·00	27·90	$7\frac{13}{16}$
No. 3	1·38	44·1	3·44 × 4·75	18·50	32·40	$7\frac{7}{8}$
No. 4	1·35	36·3	3·25 × 4·37	16·00	31·10	$7\frac{11}{16}$
No. 5	1·25	31·9	3·19 × 4·00	13·10	18·84	7
No. 6	1·25	40·0	3·44 × 4·31	17·90	36·20	$8\frac{1}{4}$
No. 7	1·25	42·1	3·50 × 4·37	20·00	27·10	$8\frac{1}{16}$
No. 8	1·18	29·9	3·19 × 3·75	13·10	18·84	7
No. 9	0·99	37·1	3·62 × 3·60	19·22	29·63	$6\frac{1}{2}$
No. 10	0·92	34·5	3·63 × 3·34	18·30	22·80	$6\frac{1}{16}$
No. 11	0·92	39·6	3·80 × 3·50	22·40	28·36	$7\frac{1}{16}$
No. 12	0·90	40·5	3·88 × 3·44	21·70	29·54	$6\frac{5}{8}$
No. 13	0·89	29·9	3·50 × 3·10	17·78	24·06	$6\frac{11}{32}$
No. 14	0·88	33·0	3·63 × 3·20	16·25	22·16	6
No. 15	0·86	32·0	3·63 × 3·10	22·90	24·05	$6\frac{11}{32}$
No. 16	0·80	40·2	4·00 × 3·20	19·95	22·16	6

to affect the comparisons. As a matter of fact, when there is a difference in the rod:crank-radius ratio there is no single reciprocating weight for the short-stroke engine, which will give the same inertia forces as that of the long-stroke example for corresponding crank angles throughout the cycle. This is because the weight depends on the ratio of the inertia factors for the two engines, and this varies slightly for different crank positions.

It can be seen from the comparisons in Table V that the reciprocating weights of short-stroke engines deviate both ways from the value W_l, but that the negative deviations predominate. That is, in most of the short-stroke engines shown, the inertia forces are lower than in the long-stroke units with which they are compared. It must be remembered, however, that the short-stroke engines listed are the latest engineering developments, while, with one exception, the long-stroke ones are of somewhat earlier design. In accordance with current design practice, the pistons of the long-stroke engines might be a little lighter.

Returning to the idea of reducing the peak bearing load of a particular engine by changing the stroke:bore ratio, and in view of the foregoing paragraph, it probably will not be too far from the truth to assume that the inertia force will remain the same for different stroke:bore ratios. If the ratio of rod-length:crank-radius remains the same, the length of the rod will vary slightly with changes in the stroke:bore ratio. Let it be assumed that the weight of the lower part of the rod, which is rotating weight, will vary directly with the length. With these assumptions, it is possible to obtain, by a few trials, the changes in stroke:bore ratio of a given engine that will provide approximately equal crank pin bearing loads at top dead centre of the induction stroke and at 18 deg past top dead centre of the expansion stroke.

Considering engine No. 1, for example, it will be found that a change from a stroke:bore ratio of 1·58:1 to a ratio of 1·35:1 will make the crank pin bearing loads at these points in the cycle about 2,440 lb, Table IV, which is not much under the maximum of 2,644 lb. But increasing the stroke:bore ratio of engine No. 14 from 0·88:1 to 1·40:1 reduces the maximum load about 1,700 lb. The stroke:bore ratios of these two hypothetical engines, which are designated as No. 1X and No. 14X, are higher than might be expected in view of the comparisons in Table V. The reason is that engines No. 1 and No. 14, from which they were derived, have very light reciprocating parts. Engine No. 15, with a stroke: bore ratio of 0·86:1, has relatively heavy reciprocating parts. Changing to a stroke:bore ratio of 1·11:1 will reduce the maximum crank pin bearing load from 3,980 lb to 2,700 lb, engine No. 15X, Table IV. Heavy reciprocating parts actually reduce the bearing load during the first part of the expansion stroke because the inertia and gas forces are opposed.

It should not be assumed that the foregoing method is offered as a means of determining the ideal stroke:bore ratio for an engine, for there are other considerations besides reducing maximum bearing load. Crankshaft rigidity is essential for the satisfactory performance of the modern high-speed, high compression engine, and the fact that the short-stroke type of construction results in a more rigid crankshaft is one of the chief reasons for the trend towards very short stroke engines. Stroke:bore ratios in the neighbourhood of unity make possible an overlap of journal and crank pin cross sections. This strengthens highly stressed portions of the shaft and tends to prevent concentration of stresses in the crank arms. The increased torsional stiffness of the shaft may raise the speed at which periodic vibration occurs to a level above the speed range of the engine. Also, the short-stroke engine, with its short crank throws and compact crankcase, is fundamentally lighter.

The short-stroke type of construction is well suited for high-speed engines, but there are no grounds for the idea that there is an inherent tendency for the short-stroke engine to develop maximum power at higher r.p.m. than the long-stroke type of unit. Ignoring small variations due, for example, to variation in mechanical efficiency, it can be assumed that an engine tends to run as fast as the size of its valves permit, regardless of the stroke:bore ratio used. This is, of course, provided there is no greater outside restriction to gas flow. But the higher the stroke: bore ratio, the higher will be the piston speed, because the small diameter piston moves further to displace the same volume of gas in a unit of time.

DYNAMIC BALANCING MACHINE

A NEW dynamic balancing machine, type 1252, has recently been developed by Dawe Instruments Ltd., 99 Uxbridge Road, London, W.5, for components up to 100 lb in weight. Essentially, the new machine works on the same principles as the established smaller 1250 machine. The component is rotated by a built-in three-phase synchronous electric motor through a flat woven belt. When the component is being rotated, any unbalance will cause vibration in the supporting structure. This vibration is detected by two moving-coil pick-ups, and alternating voltages are generated which are proportional to the magnitude of vibration at the points contacted by the pick-ups.

From a knowledge of the geometry of the machine, it is possible to develop simple relationships between the two vibration readings; they indicate the out-of-balance in any two desired planes. Electrical mixing circuits are provided to solve the relationships automatically so that the desired result is indicated without need for calculation. The mixing networks also ensure that the corrections made in one plane do not affect the readings for the other.

At the same time as the amount of unbalance is read off, the angular position of unbalance is indicated by a pointer on a numbered scale on the rotor. This scale can be formed either by permanent markings on the rotor or by a piece of numbered adhesive tape stuck in position for the test. In either case, a stroboscope is triggered by the vibration pick-ups at the point of greatest vibration, so that the scale is "frozen" at this point. In this way, the machine, and not the operator, selects the exact position, and does so in completely unambiguous terms. The stroboscopic image will indicate either a "heavy" or "light" position where material must be subtracted or added to attain balance. To prevent external vibration from affecting the results, the balancing machine is resiliently mounted.

The high selectivity of the amplifier, tuned to the running speed, attenuates external vibrations and noise, and gives a steady indication even at the smallest detectable unbalance. In addition, the assembly of pick-ups flexibly mounted on flat springs, forms a low-pass filter which effectively eliminates any vibrations that may exist in the framework at frequencies corresponding to the balancing speed.

A very high order of accuracy is maintained independent of mains voltage and frequency variations, and normal ageing of the valves and other electronic components. Negative feedback and stabilised power supplies keep the amplifier gain constant, and an automatic tuning circuit is incorporated for the filter. Thus, once the amplifier has been initially tuned to the balancing speed, it will remain in tune to provide accurate indications even though the speed varies owing to changes in the mains supply frequency.

The sensitivity of the machine may be pre-set to give the shortest possible balancing time compatible with the accuracy required. Very rapid balancing can be achieved for ordinary commercial work; slightly longer is necessary for the highest possible degree of balancing. An efficient electro-magnetic brake helps to keep balancing time to a minimum. On this machine a 100 lb rotor can be balanced to 0·0064 in oz. or to an out-of-balance force of only just over 1/6 oz at 1,000 r.p.m.

GERMAN PUBLICATIONS

The Valve Gear and Gas Flow in High Speed Internal Combustion Engines. (Die Steuerung des Gaswechsels in Schnellaufenden).

By W. D. Bensinger.

BERLIN: SPRINGER-VERLAG, GERMANY. 1955. 6×9, 93 pp., Price 12 DM.

This book is the sixteenth in a series of works on design, written by practising engineers, for designers, students and young engineers. It is by a senior engineer of the Daimler-Benz Company. The author begins with fundamental considerations of valve timing, and torsion stresses imposed on camshafts. He then deals with the basic types of valve arrangements, illustrating his work with representative engines such as the Daimler-Benz 170V, Rover 60, BMW 501, Deutz F8L614, Buick V8, Armstrong-Siddeley Sapphire, etc. These engines are critically examined with regard to the valve layout. Overhead camshaft arrangements are reviewed against the background of the BMW nine-cylinder radial and Daimler-Benz 603 aircraft engines, the Daimler-Benz 3 litre racing engine, as well as the Singer SM 1500 and Jaguar XK 120 units. This section occupies 26 pages.

In the next 20 pages, the author deals in considerable detail with the design of cam profiles. The relevant equations concerning lift, velocity and acceleration are given for a considerable number of profiles, and the method of parameter determination and cam layout is illustrated by examples. Particular attention is devoted to the layout of cam profiles for the elimination of shock; here again, the relevant determinations are illustrated by an example worked out in detail by the method developed by Kurz.

The design and stressing of valve springs is discussed in some 13 pages. In this section, spring surge and the resultant stresses could profitably be dealt with in greater detail, and some examples of anti-surge layouts adopted by different designers would be appropriate. Valve material, valve rotation and valve seats are considered next. This is followed by a discussion of camshaft stress considerations and the layout of camshaft drives with particular reference to chain drives. The remaining 33 pages of the book are devoted to porting, and rotary as well as sleeve valves. In the section on porting, two-stroke engines, including the Junkers opposed piston diesel engine for aircraft, and the Südwerke poppet exhaust valve engine are discussed. Sleeve-valve arrangements are critically examined with reference to the Knight and Burt-McCollum designs, as exemplified by the Bristol and Napier engines. It is of interest to note that for a number of reasons, the principal of which is mechanical complication, the author does not favour the sleeve-valve layout in either form. The book is concluded by a section on rotary valves. In this section the author deals with the Bristol swash-plate engine, the Sachsenberg-Sklenar engine, the Cross rotary valve and the Aspin arrangement, and disc type valves as developed by him, in conjunction with F. Wankel at the German Aeronautical Experimental Establishment.

This is a most useful book and the author has produced a clear, authoritative, unambiguous work on the design of valve gears. It will be welcomed by designers and users. As a point of criticism we feel that future editions of this excellent book would benefit by the inclusion of a chapter on rocker arm stress determination, an aspect sometimes overlooked in design.

Investigation Concerning Downhill Brakes For Heavy Road Vehicles (Untersuchung über die Talfahrtbremsen Schwerer Strassenfahrzeuge).

By Wilhelm Endres, Dr.Ing., Professor and Head of the Inst. for I.C. Engines and Vehicles at the Tech. Highschool, Munich.

DÜSSELDORF: DEUTSCHER INGENIEUR-VERLAG G.m.b.H., 1955, $8\frac{1}{4} \times 11\frac{3}{4}$, 36 pp., 37 illustrations, 5 tables. Price 14 DM.

This is the latest (No. 86) of a series of monographs dealing with research into various aspects of road vehicle design and performance, sponsored by the German Ministry of Transport, carried out mainly at universities and published by the Society of Engineers (V.D.I.). The subjects dealt with are of topical interest and the results of the investigations are usually presented in a form of direct use to designers and operators; this monograph is no exception.

Since with ever increasing loads and speeds the problem of braking heavy buses and lorries down long grades has become one of considerable importance, a thorough and authoritative treatise dealing with brakes other than conventional brakes applied directly to the wheels is rather overdue. Preliminary tests carried out by F.V.R.D.E. of the Ministry of Supply on heavy lorries show that the use of exhaust brakes appreciably improves the performance over hilly routes and reduces brake lining wear to about one-half, as compared with the one-third of the original values reported by a number of Swiss operators. Subsequently, exhaust brakes were introduced by bus operators in Wells, whilst elsewhere the replacement of trolley buses by buses drew attention to the substantial benefits derived from rheostatic or regenerative braking as far as brake lining is concerned, and this again prompted the desire for exhaust braking down grades. Although articles and papers dealing with various aspects of exhaust braking were published from time to time, both in this country and abroad, no thorough treatise was devoted to this important subject; the Enders monograph closes the gap existing between the theoretical evaluation and appreciation and the practical application of engine exhaust and some other shaft brakes.

At the outset the author considers the reasons for using brakes other than conventional wheel units and points out that among other advantages their use permits an increase of mean speed of lorries and buses down long grades of up to 15 to 20 per cent. Indirect brakes, it is pointed out, have been adopted to complement wheel units in order to meet the conditions arising from heavier weights and higher speeds, but it is stressed that improvements in wheel brakes may reduce the present importance of indirect (exhaust, etc.) brakes. The scope and limitations of indirect brakes are dealt with next on the basis of theoretical considerations, the results being presented in the form of graphs clearly showing the limits of safe operation in terms of grade versus the number of braked wheels and coefficient of wheel to road adhesion μ. The author concludes that for $\mu=0{\cdot}2$ the limiting grade will be 1 in 7·15 to 6·25 for single buses and lorries, 16·0 to 12·5 for lorries with trailers, and 1 in 25 for empty lorries with laden trailers.

Indirect brakes are divided into those applied in front of or behind the transmission. The former, represented by the action of the engine acting as a compressor, are considered next in considerable detail. To start with, the author deals with the principles of exhaust braking as applied to four-stroke engines and, in addition to modern exhaust brake valves of Haller and Z.F.-Oetiker design, considers earlier attempts to ensure braking action by the provision of additional cams on the camshafts.

The results of tests carried out on petrol and C.I. engines to determine b.m.e.p. developed as compressors are reviewed, and the more recent results due to Eiseler, Steyr and the T.H. Munich are of particular interest. These are followed by examples, worked out in detail, of exhaust brake effectiveness in terms of speed versus grade for buses and lorries pulling trailers. The use of two-stroke engines for exhaust braking is also analysed in some detail. A study of the relevant chapters is of particular interest in view of the growing popularity of these engines. Because of the obvious importance of the interaction between the normal type of multi-speed transmission and the engine used as a brake down grades, and the possible necessity of changing gears, this aspect of operation is considered in some detail. In this connection the author stresses two major points, namely, that the exhaust brake should not be considered as a second (alternative) brake as far as vehicle maintenance is concerned, and that under no circumstances should it be possible to stall the engine owing to closed exhaust at the moment when it is to be accelerated to change gears.

Brakes provided between transmission and back axle are considered next. In this section the author deals with the design and performance of the Telma eddy current brake and the Westral (a Westinghouse product) water-cooled disc brake, both types being applied to the cardan shaft.

The author has succeeded with a minimum of mathematics to present the most thorough and rigorous analysis of exhaust braking yet published, and the presentation is clear and lucid and remarkably concise. Altogether, this is a most interesting and useful publication on a timely and important subject, and its careful study can be thoroughly recommended to every designer and operator of commercial vehicles. The monograph is produced to the high standards we come to expect from the publishers, the drawings and graphs being particularly clear and self-explanatory, so that the publication can be studied with profit even by those not familiar with German.

CITROËN HYDRO-PNEUMATIC SUSPENSION

A System that Automatically Adjusts Itself to Suit the Load

IT is claimed that the hydro-pneumatic suspension employed in the current version of the Citroën Six saloon completely eliminates two common causes of passenger fatigue; they are fore-and-aft pitching on undulating roads and mechanical vibration. The new suspension is also adaptable for use as a jack to raise and lower the vehicle for changing the rear wheels. A manual control for this is inside the boot.

Work on the development of the new system has been in progress for some years. It included running the vehicle on roads in which there were 6 in deep pot-holes and on which were scattered stones the size of building bricks. The manufacturers state that, despite the flexibility of the system, the high-speed road holding qualities associated with the Citroën Six equipped with the conventional type of suspension have not been impaired.

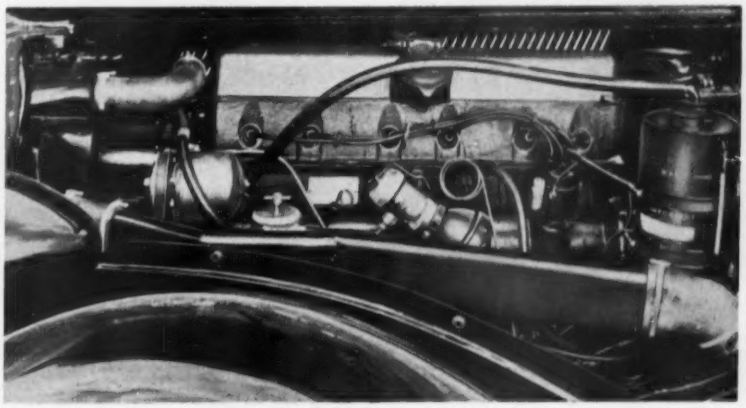


Fig. 1. The pump and hydraulic accumulator are mounted on the engine and the reservoir for the hydraulic fluid is on the dash

General arrangement

The general arrangement of the new suspension system is as follows. At the front, the torsion bar arrangement used hitherto, on the conventionally sprung car, is but little changed. The length of the torsion bars has been altered and the rate reduced slightly. In addition, an anti-roll bar, connected to the lower arms of the transverse wishbones by drop links, has been installed.

At the rear, the suspension system is mounted on a frame extending rearwards from the body sills, Figs. 2, 3 and 10. This frame is based on a steel tubular cross member bolted to the sills by means of attachment plates welded on its ends. The bolts through each attachment plate are widely spaced to react the couple due to the overhang of the frame to the rear of this tube. Two rearward extending, box-section, longitudinal side-members are welded to the tube, one near each end, and another large diameter steel tube is welded between their rear ends.

A trailing link suspension layout has been adopted. The link on each side is of bell crank form with one arm shorter than the other and extending downwards. This arm carries the bump and rebound rubbers, the movement of which is limited by stops on the trailing member of the frame. The longer arm forms the trailing link and carries the stub axle. The pivot pin is at the junction between the two arms and is carried in a bearing in the trailing member of the frame. To its inner end is attached a rectangular block, which is in two pieces. The pieces are clamped one on each side of the swaged rectangular section end of an anti-roll bar, which thus connects the pivots of the two trailing arms.

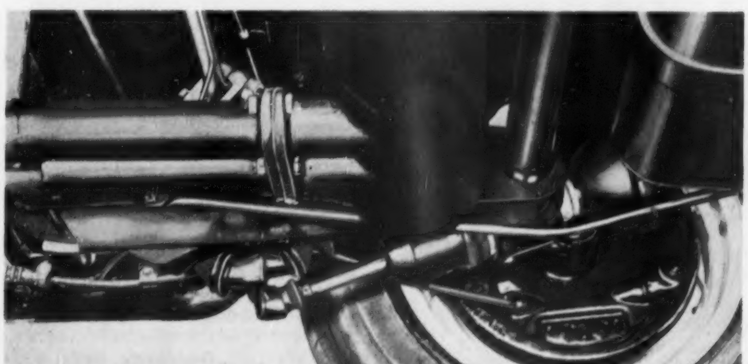


Fig. 2. The hydraulic spring units are pivot mounted on the ends of a frame cross tube and their lower ends are connected to the bump-stop arms of the trailing links of the suspension system

The spring is formed by a hydro-pneumatic jack on the outer end of which is screwed a sphere that contains the gas under compression and the damper fluid. The cylinder is pivot-mounted near the end of the trailing member of the frame, while the piston rod is attached to the lower end of the bump stop lever. Thus, as the wheel rises and falls, the hydro-pneumatic jack is compressed and extended between the bump stop arm and the trailing frame structure.

Hydro-pneumatic system

Hydraulic pressure is supplied by a pump mounted on the front of the cylinder block and driven by a V-belt from a crankshaft pulley, Fig. 1. The fluid is supplied from a reservoir mounted on the dash. Thence it is delivered to a hydro-pneumatic accumulator in which is incorporated a distributor valve that automatically causes the system to be pressurized from the accumulator when the pump is inoperative. When the pump comes into operation this distributor valve feeds the hydraulic fluid under pressure into

the accumulator to recharge it, as well as into the suspension system. From this unit, the hydraulic circuit passes through an isolation cock, to the automatic height corrector valve. A pipe connection is taken from this valve to each hydro-pneumatic spring unit. The function of the valve is to regulate the pressure in the spring units and thus to allow for variations in the load carried.

Reservoir and pump

The liquid employed in this system is the same as that used in the hydraulic brake circuit. A reservoir, in the filler of which is incorporated a filter, is carried on the dash. Mounted vertically on the side of the reservoir is a glass-tube gauge to indicate the height of the fluid, Fig. 11.

A dome-shaped cover is fitted over

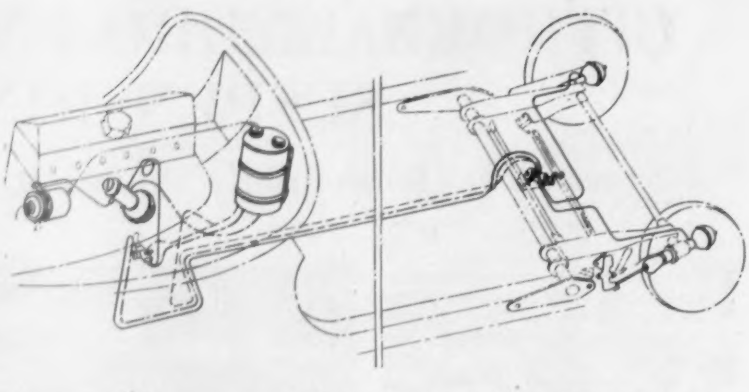


Fig. 3. Diagram showing the layout of the Citroën hydro-pneumatic rear suspension system. From left to right, the main components are: hydraulic pump, accumulator and distributor valve, isolation cock, height corrector valve, and the two spring units

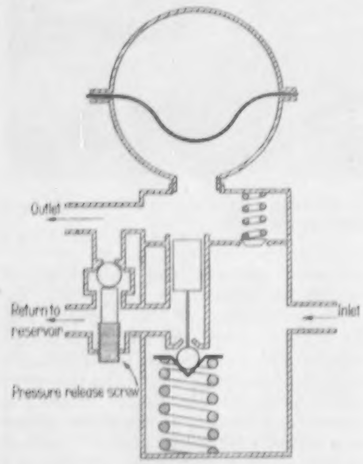


Fig. 4. Diagrammatic illustration of the essential features of the accumulator and distributor valve

the pump, which is a seven-cylinder, swash-plate type unit. The drive shaft, of course, is in the centre, with the cylinders disposed round it, parallel to its axis. On the delivery stroke, the swash plate actuates the pistons; on the return stroke they are actuated by coil springs in compression. A delivery valve of the non-return type is fitted in the end of each cylinder; the inlet port is in the side of the cylinder, and is uncovered by the piston as it moves outwards on the suction stroke.

Distributor valve and accumulator

As can be seen from Figs. 4 and 6, the distributor valve is screwed on to the accumulator. Hydraulic fluid under pressure from the pump enters an intake chamber in the valve body. Thence it passes through a non-return valve into the accumulator chamber and then to the automatic corrector valve. Pressure in the accumulator chamber acts on a piston and, when it reaches a certain value, causes the piston to unseat a spring-loaded ball valve. The unseating of this valve allows oil in the intake chamber to by-pass the accumulator and return to the reservoir. In the by-pass circuit, there is a pressure release screw which normally holds a ball valve on its seating in a passage connecting the outlet from the accumulator with the outlet to the reservoir. When this screw is slackened off, the valve is opened and releases the pressure in the system, allowing the excess fluid to pass back to the reservoir.

The accumulator, on the end of the distributor valve body, is a sphere divided diametrically by a flexible membrane in a plane normal to the axis of the distributor valve body. In the sealed half of the sphere is a gaseous mixture, normally under pressure, while the other part contains hydraulic fluid and is in communication

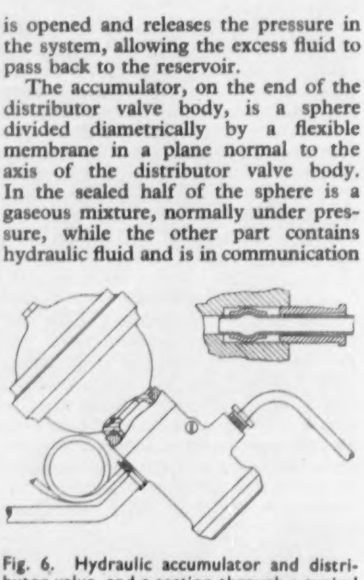


Fig. 6. Hydraulic accumulator and distributor valve, and a section through a typical pipe joint

with the accumulator chamber in the distributor valve.

Isolation cock, height corrector and hydro-pneumatic spring unit

The isolation cock is fitted in the circuit between the accumulator and the height corrector. Its function is to enable the rear portion of the circuit to be isolated from the front part, so that the system can be locked to maintain the suspension at its normal level when the engine, and therefore the hydraulic pump, is stopped. The cock is controlled from the dash, but it is opened automatically by the first application of the clutch pedal after it has been closed, Fig. 5.

An automatic height corrector valve, Fig. 7, is incorporated in the circuit between the isolation cock and the hydro-pneumatic spring units. It is simply a slide valve actuated by a tongue brazed to a rod and mounted so that it can pivot round the anti-

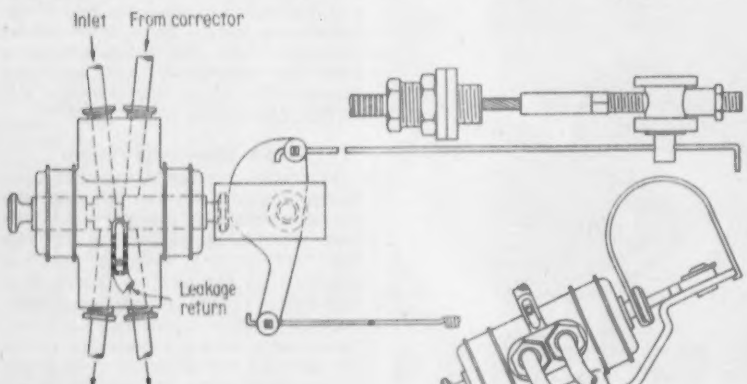


Fig. 5. Two views showing diagrammatically the general arrangement of the isolation cock

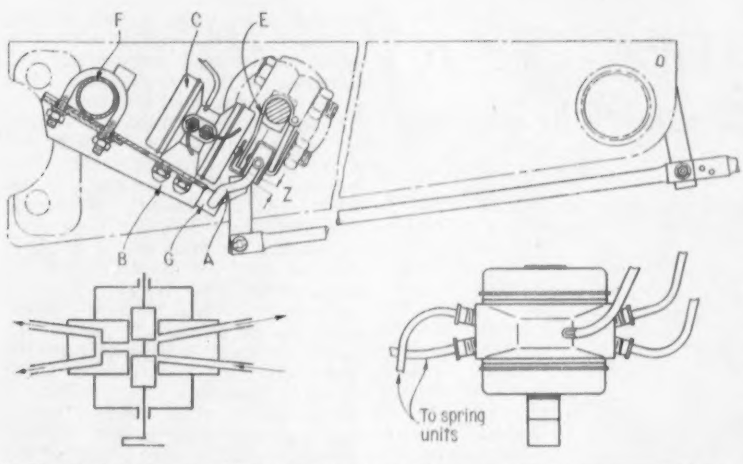


Fig. 7. Height corrector valve and its manual control layout. Left, a diagrammatic illustration showing the principle of operation of the valve

roll bar. The extremities of the rod to which it is brazed are clamped between the end fixtures of the anti-roll bar. Thus, as the suspension arms rise and fall, the slide is moved up and down by this tongue. When the arms rise, the slide uncovers the delivery port in the valve unit and allows fluid to pass under pressure into the hydro-pneumatic spring cylinders, and when the arms fall, the delivery port is closed and the fluid allowed to pass from the spring units through a return port to the reservoir. So long as the suspension remains at the normal height both the delivery and return ports are closed.

Another control is attached to the slide valve, and it can be operated manually by a lever in the boot. This control makes possible the use of the suspension system as a jack so that the car can be placed on a stand for wheel changing. The stand is inserted under the vehicle on one side or the other immediately in front of the rear wheel. Then the control is placed in the "low" position to release the suspension arm so that the wheel can be removed.

As has already been stated, the hydro-pneumatic spring unit, Fig. 8, consists essentially of a sphere screwed into the end of a jack cylinder. The gaseous mixture under compression in the spherical end is separated from the hydraulic damper fluid by a flexible membrane. Damper valves are carried in the neck of the spherical component, where it is screwed into the cylinder. The other end of the unit is closed by the piston of the jack, and a flexible gaiter protects it against the entry of foreign matter. There is only one pipe between each spring unit and the corrector valve. This pipe, of course, serves both as the pressure and return line.

Height adjustment during service

Adjustment to the height of the rear axle is effected on a level, horizontal surface, after the tyre pressures have been checked and the height of the front suspension adjusted, but before the two shock absorbers are connected. With the engine ticking over, the wheel-change control lever in the boot is placed in the "low" position. When the car has settled down, the rebound rubber is removed from the arm on one side and a special stop is mounted in place of it.

Next, the wheel-change control lever is moved to the "high" position. When the vehicle has stopped rising, the left-hand side is lifted still further with a jack and a block is placed underneath it for safety. Although the vehicle should be raised as high as possible to give easy access to the height corrector, the right-hand wheel should nevertheless remain on the ground. The next operation is to remove the bump rubber from the second suspension arm and replace it by a stop, similar to the one previously fitted in place of the rebound rubber on the other side.

Then the engine is stopped and the pressure in the system released in the following manner. Check that the isolation cock is open and place the wheel-change control lever in the "low" position. Then, close the cock and make sure, by actuating the suspension units by hand, that no pressure remains in the rear circuit. The units should be free in their supports. Return the wheel-change control lever to the "normal" position. Then uncouple the wheel-change control rod A, in Figs. 7 and 9, from the

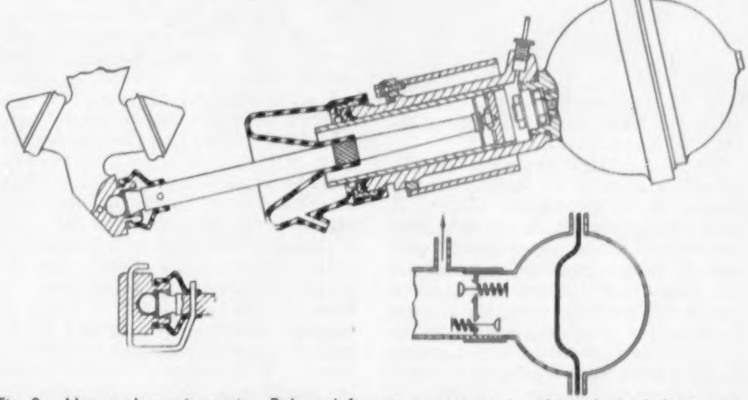


Fig. 8. Above: the spring unit. Below: left, a transverse section through the ball joint at the lower end; right, a diagrammatic illustration showing a section through the sphere and indicating the function of the damper valve

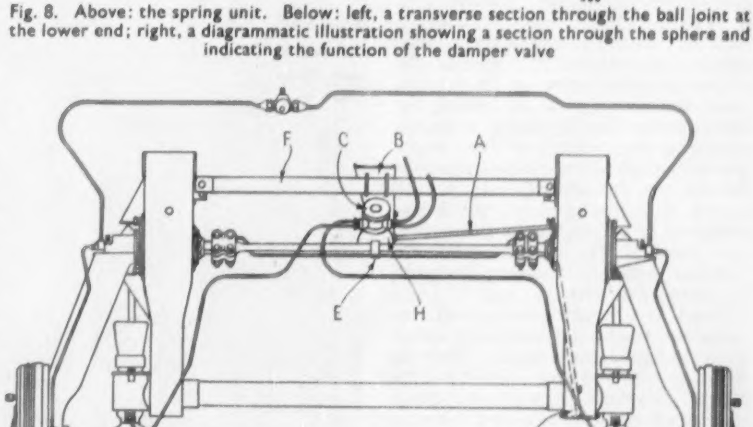


Fig. 9. Among the mechanical components of the hydro-pneumatic rear suspension system is an anti-roll bar that connects the pivots of the two trailing arms

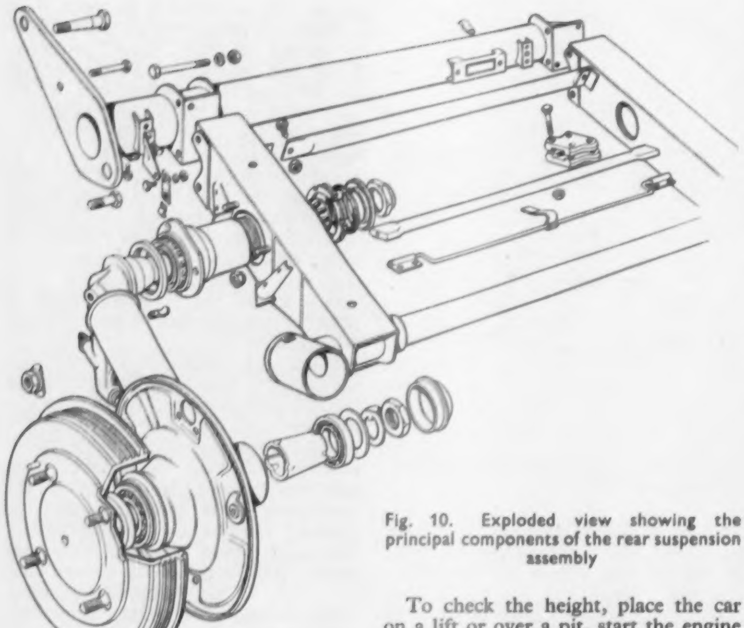


Fig. 10. Exploded view showing the principal components of the rear suspension assembly

corrector bracket, B, and disengage it from the yoke on the height corrector valve, C. Next, remove the end of the rod from the pivot-support, D.

To set the height corrector approximately, centre it in relation to the tongue, E, by moving the bracket, B, on the cross tube, F. At the same time, the yoke of the height corrector valve and the tongue should be parallel to one another; if they are not, rectification is effected by rotating the bracket, B, about the cross member, F. Adjust the depth of engagement of the tongue, E, by moving the cross tube, F, on its end supports until the dimension, Z, is 4 mm. The clearance between each side of the tongue and the yoke should be equalized by moving the height corrector valve on its bracket. This operation should be carried out by sighting, not by using a gauge, otherwise the valve of the height corrector might be moved accidentally. Couple up the control rod, A, and centre it in the yoke of the height corrector by adjusting the position of the bearing, G, on the corrector bracket. Finally, the rubber protector, H, is placed in position.

Start the engine and open the isolation cock by depressing the clutch pedal and releasing it again. Place the wheel-change control lever in the "high" position. Replace the bump rubber after having removed the stop. Lower the car to the ground and put the control lever in the "low" position. Fit the rebound rubber in place of the stop. Insert a screwdriver between the bump rubber and the cup to release any air that might be trapped between the two.

To check the height, place the car on a lift or over a pit, start the engine and allow it to tick over. Chock the front wheels and release the hand brake. Put the wheel-change control lever in the "low" position and wait for the car to settle. Next, move the control lever to the "normal" position and again wait for the car to settle. Remove the wheel-change control rod, A, and the bearing, G. Place a straight-edge over the pit or on the car-lift, so that it is parallel to the rear, tubular cross member of the car and its lower edge is in the same plane as the areas of contact of the tyres with the ground or lift. It may be necessary to make a special straight-edge for this purpose. Raise the car very slowly by lifting at the rear bumper. Stop when a resistance is felt and wait until a whistle is heard. This indicates that the liquid is completely exhausted. At the instant the whistle is heard, a second operator must make a note of the dimension, x_1, between the underside of the tubular cross member and the underside of the straight edge. Let go the car and wait for it to settle.

Next, push down on the rear bumper until a resistance is felt. Wait in this position until a whistle is heard. This indicates the end of the intake of the liquid. At this instant, the second operator should note the dimension, x_2, between the underside of the cross member and the underside of the straight-edge. Then $(x_2+x_1)\div 2$ should be between 269 and 285 mm. Adjustment can be effected by moving the height corrector. A displacement of the height corrector a distance of 1 mm upwards increases the height of the vehicle by about 8 mm; a corresponding movement in the opposite direction reduces the height of the vehicle by about the same amount. When adjustment is complete, the wheel-change control rod is coupled up and centred in the yoke of the height corrector. Finally, the rubber protector is placed in position and the front shock absorbers connected.

Conclusion

This suspension incorporates a number of commendable features. One is its arrangement on a trailing frame. This allows the concentration of all the structural components of the body structure into an area forward of the rear ends of the sills. Thus, the rear end of the body can be made lighter than would otherwise be possible. This presumably would be a greater advantage on a car of conventional layout than on a front-wheel-drive vehicle.

Another advantage of this system is the elimination of vibration troubles and fatigue failures. Apart from this the automatic adjustment for variation in static load is a most desirable feature. The facility with which wheel changing can be effected, no doubt, will also be appreciated by owners of the Citroën Six equipped with this suspension.

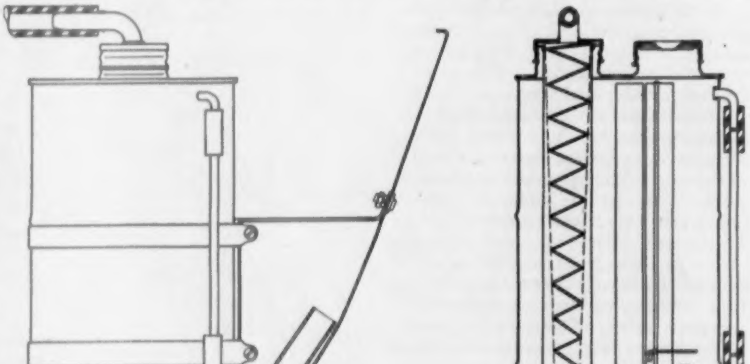


Fig. 11. A glass tube is mounted vertically on the side of the reservoir to indicate the height of the fluid

NEW PLANT AND TOOLS

Recent Developments in Production Equipment

A NEW lapping machine, known as the Mark IV, manufactured by Flexibox Ltd., Nash Road, Trafford Park, Manchester, 17, is shown in Fig. 1. It is designed to lap components ranging from non-metallic soft materals to the very hard metals, and is claimed to be considerably lower in price than any comparable machine. A rotary lapping plate of 21 in diameter gives high loading capacity with correspondingly high output rates. For example, 99 workpieces, $\frac{3}{4}$ in diameter, can be lapped simultaneously. The work to be lapped is held in three wear rings and, in general, any components that can be accommodated within a circle $7\frac{3}{4}$ in diameter can be lapped.

The machine is fitted with a control device which automatically maintains the flatness of the lapping plate. Only one simple adjustment of the device is necessary. Thereafter, the plate is continuously conditioned during operation. Loading and unloading are extremely simple, and the workpieces are protected by a transparent plastics hood. Once the control device has been correctly set, the machine can be operated by unskilled labour. If reasonable care is taken, the largest parts for which the machine is suitable can be lapped to within one light band and two micro-inches R.M.S. surface finish.

Fig. 1. Mark IV Lapping Machine
(*Flexibox Ltd.*)

Six-spindle bar automatic

A new six-spindle bar automatic recently added to the range of multi-spindle machines manufactured by Wickman Limited, Banner Lane, Coventry, is shown in Fig. 2. It is designed to take round bar up to 2 in diameter, and in common with all machines in the Wickman multi-spindle range incorporates the patented auto-setting mechanisms which allow alterations to tool feed strokes to be made without any cam changing.

The spindles are driven from a centre shaft and are mounted in taper roller bearings at the front and parallel roller bearings at the rear. Labyrinth and piston ring seals are provided to exclude coolant. The collets are closed by toggles with a pre-loaded compensating device to allow for variations in bar size, while maintaining gripping power and ample opening movement to allow the bar to feed freely.

To maintain accuracy, the spindle drum over-indexes past the latch and is then drawn back by toggle mechanism. The guide of the main block indexes with the drum. This ensures permanent accuracy. End float of the drum is closely controlled and is not subject to variation due to temperature differences.

Fig. 2. 2 in six-spindle bar automatic
(*Wickman Ltd.*)

A very long main block is employed. This provides greatly increased space for tooling and attachments. It is fitted with piston ring seals and cast iron bushes. The independent slides are carried on wide dovetails. They are provided with safety stops. Heat-treated steel is used for the cross slides, which have a much improved range of strokes. All have micrometer adjustment, adjustment locks and master stops. Six slides are available, and for special tooling either the intermediate or the upper cross slide on each side of the machine can be timed independently. All slides have infinitely variable feed stroke from zero upwards.

The gear box provides a wide range of feeds and speeds. It has large multi-disc clutches, as well as safety clutches and trip mechanisms. All the controls are duplicated at front and rear, and a lever is provided for manual engagement of the fast motion clutch in such a way that it cannot be engaged accidentally. The hand crank is interlocked to avoid danger to the operator. There is a large attachment drive compartment, and all the attachment drives have been specially

Fig. 3. No. 0 centreless grinder
(Cincinnati Milling Machines Ltd.)

designed for easy building-in as units in the shortest possible time.

Bar feed is by cam-controlled springs with a special safety device incorporated in the return mechanism. The length of waste bar is constant irrespective of stroke. Manual collet operation is provided with ample leverage for easy operation. The bar stop is adjustable for length of component and swings down into position, thus remaining clear of swarf. An auto-stop device is available to trip the feed at a point where the collet is open when the stock in any spindle is depleted.

A motor driven swarf conveyor is available. It is recommended that this device should be fitted to every machine. This conveyor can be conveniently plugged in or removed for cleaning. It is protected against damage by a magnetic starter and a shear pin device in the drive. Attachments for diehead screwing or tapping can be fitted in four stations and for high speed drilling in all stations. Double bar feed, that is, in two stations with single index, is also available. Machines so equipped are fitted with automatic stock depletion trips in both stations.

Centreless grinder

A new small, general purpose centreless grinding machine developed by Cincinnati Milling Machines Ltd., Woodlands Farm Road, Tyburn, Birmingham, 24, is illustrated in Fig. 3. It is designated the No. 0 and has been specially designed to deal with metallic and non-metallic parts up to ½ in diameter. The grinding wheel spindle is mounted in a Cincinnati Filmatic spindle bearing. This is a multiple segment bearing that supports the spindle accurately and rigidly on high pressure, wedge shaped oil films. The bearings are self-adjusting for heavy roughing or light finishing cuts, and do not require any maintenance for the service life of the machine. Spindle lubrication is automatic and positive. An automatic cut-out is provided for stopping the spindle motor if the lubricant should fall below a specified level. The spindle bearings are rigidly mounted in the bed casting, without joints or sliding elements. This construction assures the maximum rigidity for rapid and accurate metal removal and fine finishes.

The regulating wheel unit is carried on the base in two wide dovetailed slides to permit adjustment of the work-rest in relation to the regulating wheel, and adjustment of both work-rest and regulating wheel in relation to the grinding wheel. To permit correction of slight errors in straightness without having to re-true the wheels, a swivel plate is located between the lower slide and the base.

Under the control of a single handwheel, the regulating wheel speeds are infinitely variable from 22 to 300 r.p.m. This allows the correct speed to be used for efficient cutting on any particular work. It is also useful for establishing exact production rates when machines are used in tandem. The regulating wheel speed is indicated on a tachometer mounted at the rear of the regulating wheel spindle. Adjustments of the regulating wheel unit are effected by means of a pilot-type handwheel fitted with a large diameter micrometer dial. A conveniently placed hand lever permits rapid grinding by the infeed method.

The standard grinding wheel truing attachment is hydraulically actuated and trues the wheel either to a straight cylindrical shape or, if desired, to slight tapers. A profile hydraulic truing attachment for formed profiles or multiple diameters is available at extra cost. The standard regulating wheel truing attachment is of the straight screw-type, but a profile screw-type, a straight hydraulic-type, and a profile hydraulic-type are available at extra cost. All truing attachments are of the diamond type, and the profile truing attachments are equipped to take either round or flat profile cams. Work-rests are available for both infeed and through-feed grinding.

The grinding wheels are 16 in maximum diameter, with 10 in diameter hole and 4 in maximum width; the comparable figures for the regulating wheel are 9 in, 4 in and 4 in. A separate cutting fluid tank with a motor driven cutting fluid pump of 21 gallons per minute capacity is located at the rear of the machine.

Strip former

To increase the range of components for which four-slide strip forming machines can be used, Heenan and Froude Limited, Worcester, have developed a larger and more powerful machine—the S.207—which is illustrated in Fig. 4. It is thought to be the largest and most powerful machine of its kind. An important characteristic of the new machine is that a very long die set can be used, some 21 inches on the standard machine and 31½ inches on the special. This enables more progressions to be obtained than is usual with the general range of strip formers. Consequently, it is possible to produce more difficult forms.

The standard machine incorporates an accurate mechanism for feeding and straightening the stock, and is available in the following forms:—

Fig. 4. S.207 four-slide strip forming machine
(Heenan and Froude Ltd.)

(1) Fitted with two rams, each of 15 tons capacity, a stock shearing head and four forming slides.
(2) Fitted with one 15 tons capacity ram, a toggle press unit of 100 tons capacity with stock shearing head, and four forming slides.
(3) Fitted with three 15 tons ram units but without a stock shearing head and left-hand slide.

A five-roll strip straightener is incorporated in the machine. It has adjusting rolls and guide rolls in both planes for strip entry, and a quick release mechanism. The stock is automatically held by a positive grip. A maximum length of 14½ in of strip can be fed per stroke. For setting the feed there is an adjustable crank disc which is connected to the feed slide by links and levers. When the correct length of feed has been set, the crank pin can be locked. The stationary grip is controlled by a cam, which is easily adjustable for timing purposes.

With the standard 15 tons maximum capacity ram, strip material to a maximum of 2½ in width and $\frac{3}{32}$ in thickness can be worked. Actually, the material thickness that can be formed successfully to a large degree depends upon the load required for the press operation. The punch block and the die block are kept in correct relation to each other by guide pins, so that they make a complete die set, which can be easily removed. The die set is 10⅛ in long on each ram. Provision is made for adjusting the shearing head along the bed to suit the component. It can be adapted either for a straight shear or, within certain limits, a shaped end can be cut according to the type of component.

Four forming slides are provided. The end of each slide is fitted with hardened steel rollers mounted on a needle roller assembly. Each slide is fitted with an adjustable tool block. Wear strips are provided for taking up play between the slides and the slideways. The maximum strokes obtainable are: front slide 2¾ in, back slide 2¾ in, side slides 4 in.

The cams are made of case-hardened steel. They are split for each change-over to give a different stroke, dwell, etc., and are adjustable for timing purposes. There is 3½ in adjustment for the centre tool head, and as the centre tool itself is mounted in a slide the blank can be transferred, if desirable, by the action of the front tool to a position more convenient to the side slides for the forming action. The stripper is operated by levers actuated by a positive-action cam. A certain amount of forming can be carried out with this. The stripper stroke is adjustable between 1¾ in and 3½ in. A 7½ h.p. motor drives the machine. Change gears are provided to give camshaft speeds of 40, 52, 65 and 85 r.p.m. A heavy flywheel is mounted on the speed shaft. A limit switch is provided so that should the machine deliver a short feed, the operating sequence is interrupted and the clutch withdrawn, thus allowing the machine to come to rest.

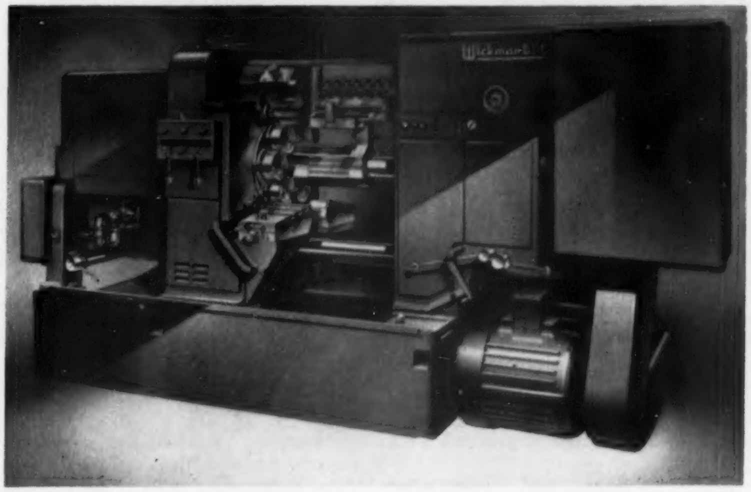


Fig. 5. 6⅝ in six-spindle chucking automatic
(Wickman Ltd.)

Chucking automatic

In addition to the six-spindle bar automatic mentioned earlier in these notes, Wickman Limited, Banner Lane, Coventry, have also developed a 6⅝ in six-spindle chucking automatic. Basically, the machines are similar and differ only in the mechanisms necessary for their specific functions. The chucking machine is shown in Fig. 5. The spindles are designed with large integral flanges for mounting the chucks and run in extra precision taper and parallel roller bearings effectively protected against swarf. Each spindle is fitted with a large multi-plate clutch and brake, operated by an individual fork. The free-runing driving gear is mounted on extra precision anti-friction bearings. Flanged adaptors are mounted on the rear end of the spindles and the air cylinders are attached by studs.

Tandem type chucking cylinders with a long draw bar stroke are employed. Each cylinder is provided with its own combined running joint and valve of an entirely new type, designed to give long life and light operation. Only one air hose is connected to each running joint and valve, and only one sealing leather is in rotating contact.

A simple reversing valve fitted to each cylinder enables changeover from external to internal gripping to be performed in a few seconds. It also allows any chuck to be opened at any station if necessary. An automatic water trap and drain, a lubricator, a pressure gauge and regulator, and a pressure safety switch are provided. The safety pressure switch is arranged to trip the feed mechanism if the air pressure falls below a pre-set minimum safety value, and to prevent the starting of the motor if the pressure is insufficient.

Fig. 6. Small bore gauge
(J. E. Baty and Co. Ltd.)

Both three-jaw and two-jaw wedge pattern chucks are available. They have 100-tons alloy steel bodies and interchangeable hardened base jaws and wedges, with optional wedge angles to provide the greatest gripping power or the greatest movement. Special chucks and workholding fixtures can be supplied to suit specific components. The control levers for chuck and clutch operation are conveniently placed and are light in operation. They electrically control air cylinders which operate the spindle clutch, brake and chuck. Interlocks

are provided to ensure correct sequencing and full protection.

Loading is performed right-handed at station 6. Indexing is anti-clockwise so that the heavier roughing operations are carried out on the lower stations. The chucks can be left open for two-handed loading of the work, and magazine loading can be arranged. Double-indexing can be specially arranged during construction of the machine, with loading to be carried out at stations 5 and 6, with individual chucking controls at each station. Either separate or common clutch control can be arranged.

Small bore gauges

To their established range of bore measuring instruments, J. E. Baty and Co. Ltd., 39 Victoria Street, London, S.W.1, have recently added a small hole gauge with a capacity of ⅛ in-½ in. The construction of a small bore indicating gauge that will repeat to 0·0001 in, presents problems owing to the confined space in which the instrument must work. Not only must the gauge incorporate a satisfactory means of transmitting contact plunger movement to the indicator, but it must also embody an arrangement for centralizing itself in the bore. These difficulties have been successfully overcome in the new Baty instruments.

The new gauge, which is illustrated in Fig. 6, is available in two forms, with straight or right-angle holders, and with indicators reading to 0·0005 in., 0·0001 in or 0·01 mm. There are two sizes of straight holder gauges; one for bores 1½ in deep and the other for bores 2½ in deep. The right-angle holder is suitable for bores 1 in deep and is designed for operation where space in front of the hole is limited. It has an average height of 5½ in and requires a space of only 2 in for insertion in the bore.

These gauges will read accurately to 0·0001 in. They are purely mechanical, with no dependence upon air or electricity, and are simple, positive and reliable. The mechanism is robust, yet responds freely under light contact pressure and has the sensitivity that is essential for accurate measurement. As with other Baty bore gauges, a shoe, independent of the measuring contacts, centralizes the gauge in a bore. There are no expensive interchangeable measuring heads, the range being obtained by extension rods.

CONVEYER DRIVE SEAM WELDER

An Interesting Sciaky Machine

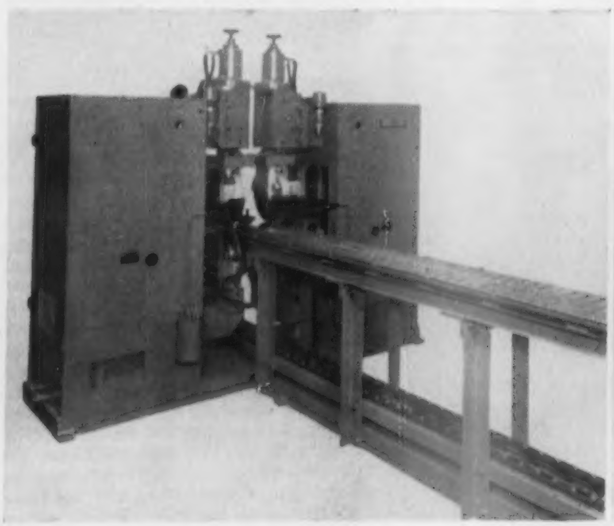
SO far as the actual welding is concerned, there are already machines that have a completely automatic cycle, but with the advent of automation there will, no doubt, soon be a demand for machines that are specially designed for inclusion in a fully mechanized production line and for working without operators. This demand has been anticipated by Sciaky Electric Welding Machines Limited, Farnham Road, Slough, Bucks., with the conveyor drive seam welding machine, RAMC 150/8551, shown in the accompanying illustration.

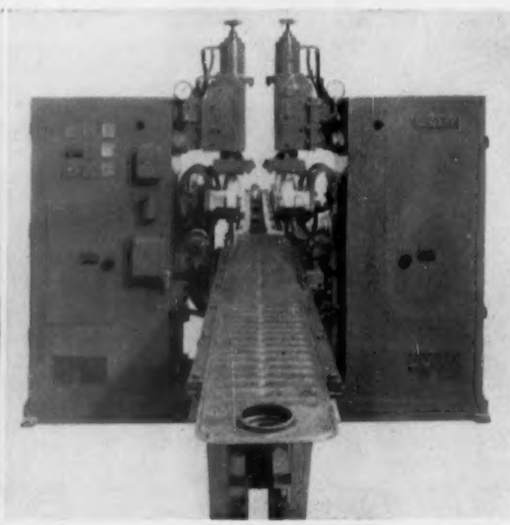
The machine has been specially designed for producing two parallel seams simultaneously on long flat components such as radiator sections. By this means, a high speed of component production is obtained, handling is virtually eliminated. It is claimed that distortion, which often accompanies the making of separate seams on a long, flat component, is also eliminated.

As is shown in the illustration, two welding units are mounted facing each other on a common base plate, with a central conveyor and guide assembly arranged between the welding units. The right-hand unit incorporates only its transformer and air pressure unit. The left-hand unit, in addition to its transformer and air pressure system, includes the conveyor motion initiation and speed control, together with the controls for the welding current initiation and pressure systems for both units.

From the pressing of the conveyor starting push-button, the machine is fully automatic in operation. The component is placed on the loading end and starts its own seam welds automatically as soon as it arrives directly between the electrode wheels. As the component comes into position, it brings both the top welding wheels down to its welding face and then initiates its own welding current. On reaching the end of the seam, the component switches off the welding current and lifts the electrode wheels clear of the welded face. It is automatically ejected at the discharge end of the conveyor.

The transformer power rating is 150 kVA at 90 per cent duty cycle. Welding pressure is adjustable up to a maximum of 1,000 lb and the speed is variable between three and nine feet per minute. The current for welding is under fully synchronous ignitron control with phase-shift heat regulation.





Two views of the Sciaky drive seam welder

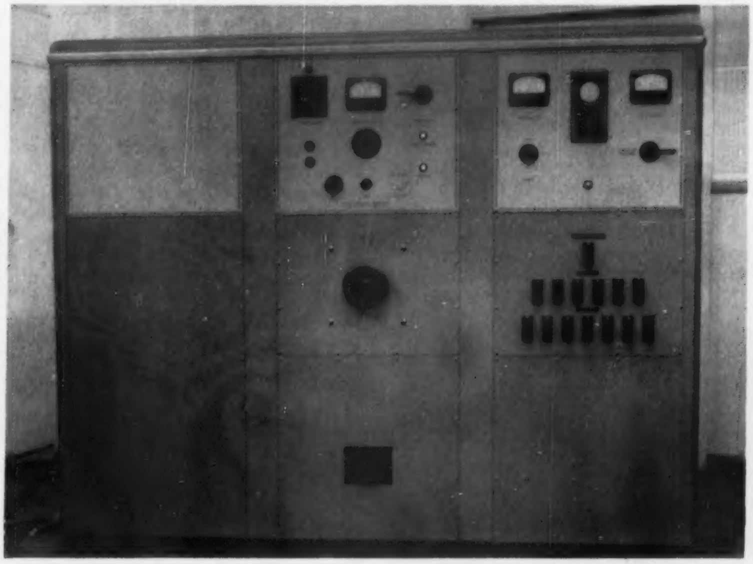
RAPID ANALYSIS OF STEEL

The Quantometer Used as a Production Control Instrument

STRICT technical control of steelmaking at all stages of manufacture and processing is of primary importance. Accurate analyses of bath samples are required during refining, and the more rapidly they can be produced the more efficient is the control. Additionally, any reduction in the time required to take a sample, make an analysis, and transmit the data back to the melting shop will improve the rate of production, be of economic value by shortening the period during which large masses of metal must be held at a high temperature, and make possible a greater utilization of plant and equipment.

At the Ickles works of Steel, Peech and Tozer, Rotherham, a branch of the United Steel Companies, Ltd., Sheffield, a Quantometer for the rapid spectrographic analysis of steels and certain other materials is in service as a production control instrument. At present there are only a few of these elaborate and costly instruments in Britain and in the main they are operated by research organizations or engaged on research programmes. This is the first instance of a Quantometer being installed in a British steelworks and it is now being used, nearly 24 hours per day, on production analyses. Although the actual analysis of a steel sample can be made in one minute, the accuracy of the result compares favourably with that obtained by the slower, traditional methods. From the despatch of a sample from the melting shop to the receipt by the melting shop of the analysis occupies less than ten minutes.

The Quantometer was supplied by the European branch, in Lausanne, Switzerland, of Applied Research Laboratories, Glendale, California. Standardized components are used throughout its construction but each instrument is designed, adjusted and calibrated to meet the specific requirements of the particular industry or laboratory. This installation covers the analysis of plain carbon steels for residual elements; alloy steels of the nickel, chromium, molybdenum type; blast-furnace sinters and various types of steelmaking slags. The constituents selected for analysis are aluminium, boron, chromium, copper, lead, manganese, molybdenum, nickel, phosphorus, silicon, tin, titanium, and vanadium in steels; and fluorine, iron, lime, manganese oxide, magnesia alumina, phosphorus pentoxide, and silica in slags or sinters.



The source unit enables electrical conditions to be selected to give a stabilized discharge to the sample

Description of instrument

Three grouped assemblies comprise the Quantometer; a source unit, a spectrometer and a recording console. The source unit supplies power to the spark gap of the spectrometer. Provision is made for varying the electrical parameters of the discharge to give "spark-like" or "arc-like" excitation.

The spectrometer consists of a 1·5 m grating instrument in which the Rowland circle is mounted vertically. This covers a spectrum range of 2,000-7,000Å with a dispersion of 6·95Å/mm in the first order and 3·5Å/mm in the second order. The diffraction grating is ruled 24,400 lines to the inch. Also included in this unit are the slit frames, mirrors, and multiplier photo-tubes.

The recording console contains the measuring circuits, power supplies, attenuators controlling the photo-tube sensitivities, automatic timers, and switching mechanisms. It controls the

In the spectrometer the light emitted by the discharge on the sample forms a spectrum. Each element beam is directed to its specific photo-cell

sequence of operation in the various steps of analysis. A Leeds and Northrup "Speedomax" recorder is incorporated, which registers each element in a group in sequence at 2·6 sec intervals.

A Schindler motor-generator set is provided to ensure stability of voltage and frequency of the power supply to the source unit. Also available in case of emergency is a Zenith stabilizer on the mains. The complete installation is accommodated in a separate room equipped with a special air-conditioning plant, supplied by the Pressed Steel Co. Ltd. Temperature is maintained at 70 deg ± 2 deg F, and relative humidity at 50 per cent ± 5 per cent.

Operational principles

The sample under analysis is excited, using the sample itself as the electrode with a counter electrode of pure graphite. Normal conditions are a measured gap of 3 mm, a pre-integration time of 5 sec and an integration time of 20 sec. Preselected lines in the resultant spectrum are isolated by the respective slits and focused by mirrors on to multiplier photo-tubes. The photo-currents in each case charge a condenser with a very high dielectric resistance.

As an internal standard, a pure iron of known composition is used. The rise of voltage of this standard iron line is traced as the exposure proceeds and at full scale deflection the exposure is automatically terminated. Simultaneously, the charging of all condensers ceases and a stepping switch selects the remaining condensers in sequence. The charge on each condenser is recorded as a step on the chart, and the height of the step represents the concentration ratio of the specific element concerned.

The instrument is pre-calibrated by an adequate series of homogeneous and accurately analysed samples covering the ranges of concentration desired. These are produced by normal chemical analysis, and working curves are prepared relating recorded readings on the Quantometer to percentage concentration.

Relative accuracy

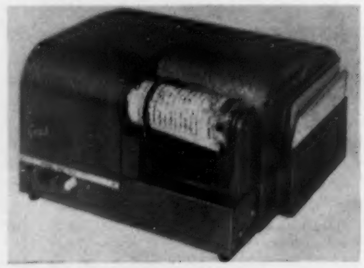
As a photometric device, using a constant light source, the instrument has been shown to have a reproducibility of better than 0·2 per cent. In routine use for residual elements up to 0·5 per cent the accuracy is better than 5·0 per cent of the concentration. For low alloy steels of up to 5·0 per cent concentration the accuracy is better than 2·5 per cent. In a research programme on highly alloyed steels the accuracy was better than 1·0 per cent of element concentration. The results obtained with the instrument will be seen, therefore, to compare very favourably with precision wet chemical analysis.

Application to production

For production control analyses in the steelworks, a sample is taken from each charge at the melt and despatched to the laboratory by pneumatic tube. On receipt there, the sample is prepared by grinding one face on a horizontal grinder and finishing on a band facer. It is placed in position on the arc-spark stand of the spectrometer unit and analysed for the necessary constituent elements. The charted records are read off and transposed to percentages with the aid of the prepared curves for the particular elements involved. These percentages are written in a standardized sequence on a form and, by a small electronic instrument, the Desk-Fax, transmitted back to the melting shop in facsimile.

The electrical circuits for measuring and recording the photo-cell currents received from the spectrometer are grouped in the recording console

To transfer the sample to the laboratory takes about four minutes, preparation of the sample two minutes, the actual analysis only one minute, and transmission of the result less than two minutes. The total time expended on the complete operation is thus only about nine minutes.



Desk-Fax instrument for the facsimile communication of analyses

Facsimile message communication

To meet the need for the rapid communication of the results of the analysis and at the same time to eliminate completely any possibility of misunderstanding, error or loss of information, the Desk-Fax electronic, two-way, facsimile system developed by Creed and Co. Ltd., of Croydon, has been installed. Measuring 12 in × 12 in × 7 in and weighing approximately 30 lb, these desk-type instruments connected by a single pair of line wires can send or receive as required, the change-over being effected by push-button control.

The message is written or typed on a standard blank of ordinary white paper, which is then wrapped round the instrument cylinder and secured by a spring ring. Operation of the "send" button causes the cylinder to rotate and the buzzer on the receiving instrument to sound. The distant operator wraps a recording blank on the receiving instrument cylinder and presses the "receive" button. Automatically, the cylinders are brought into phase for position and their speeds synchronized.

At the transmitting end, a beam of light scans the message, in a helix pitched at 125 lines per inch, as the cylinder rotates and traverses, and reflected light is focused on a photo-electric cell. The system includes a parallel balancing beam to correct optical inversion, and the amplified signals are applied to a 0·008 in diameter tungsten stylus on the instrument at the receiving end. This stylus scans the recording blank, which is a black, carbon-impregnated paper coated on one side with a grey, insulating pigment and on the other side with a conductive metallic film. The metallic backing is in contact with the cylinder, which is earthed, and the incoming signals are applied by the stylus across the paper. These burn off the pigment, exposing the black paper beneath, to reproduce the message in facsimile.

MARLES POWER ASSISTED STEERING

A Unit That is Remarkable for its Compactness

INTEREST in power steering and power assisted steering systems has recently intensified. To meet the growing demand, the Adamant Engineering Co. of Luton have introduced the Marles power assisted steering system. This system comprises five units. They are: the pump, flow control valve, the reservoir, which contains the filter, the non-return by-pass valve, and the steering unit, which incorporates the conventional cam-and-double-roller steering gear, the valves, cylinders and the pistons. In some applications, more than one of these components may be incorporated in a single housing. For example, the reservoir and filter may be housed in a single casting that contains both the pump and the flow control valve. Alternatively, the non-return by-pass valve may be built into the steering unit.

Pump and flow-control valve

A positive displacement, gear type pump is employed. Its size is determined by the need for the delivery of a certain minimum quantity of oil when the engine is idling, but at peak r.p.m. the flow of oil may be as much as 10 gal/min, which is far more than is necessary or desirable. If all this oil were to be forced through the small valve openings in the steering unit, the power losses and oil temperatures would be unacceptably high.

The oil from the pump is passed to a flow-control valve, which prevents the back-pressure from becoming unduly high. In this valve, the main passage from the pump to the steering unit is obstructed by a piston valve, the end of which is pierced by a calibrated metering orifice. A compression spring tends to hold the valve in the closed position, in which the upstream end of the piston covers the release ports in the main passageway. These ports are connected back to the pump suction line. When the flow through the orifice exceeds a certain minimum quantity, the difference in pressure between the two sides of the piston valve is sufficient to overcome the compression in the closing spring.

In these circumstances, the valve moves in a downstream direction and uncovers the ports. This allows surplus oil to be returned to the suction side of the pump. The flow-control valve is sensitive only to the flow of oil to the steering unit, and its operation is independent of the pressure in the system as a whole. It is actuated by the drop in pressure across the calibrated orifice—the pressure required to force oil through an orifice increases as the square of the flow.

A pressure relief valve is also incorporated in the flow-control unit. This serves to protect the system against excessive pressure and to limit the maximum steering effort. It becomes inoperative if the pressure falls to a value below that required to give the maximum steering effort. In most installations, the maximum pressure is about 650 lb/in^2, but it can be increased to as much as 1,000 lb/in^2, if necessary, to suit any particular application.

The steering unit itself consists of the well known Marles cam-and-double-roller steering gear, actuated partly by manual effort and partly by hydraulic pressure, regulated by a valve unit and acting on two cylinders in each of which is a single-acting piston connected to the power arm on the rocker shaft. The steering column is divided into three sections—this is the main difference between the mechanical components of the manually operated and the power assisted layouts. The upper and centre sections are connected by a flexible coupling, and a pair of spur gears is interposed between the centre section and the section in the cam gear.

Parallel to the cam, but on the other side of the rocker shaft, is the power piston and cylinder assembly. This comprises two single-acting pistons that operate on a roller carried on needle type bearings in the jaws of the power arm, which is forged on the rocker shaft opposite the jaws that carry the double-roller.

Between the mechanical steering gear and the steering column is the valve box. This box contains the centre section of the steering column, which is termed the valve shaft. The upper end of this shaft is carried in a spherical bearing, and a spur gear is mounted on its lower end. Adjacent to the spur wheel is the valve block, which is on a needle roller bearing round the shaft.

Of the four valves in this unit, two are for distribution and two for reaction. They are arranged so that the valves of each pair are axially in line, one on each side of the block. The end of the block bears against a screw in the valve box; this screw is used for adjusting the mesh of the two spur wheels. In the neutral position, all the valves are open, and the oil circulates through the distribution valves to the cylinders and then back through the reaction valves to the inside of the valve box. From there, it goes by the return line to the filter and reservoir. If a torque is applied to the steering wheel, the spur gear on the valve shaft moves tangentially relative to the other gear. This causes one valve of each pair to open further and the other of each pair to close.

The common axis of the opposed pair of distribution valves, each of

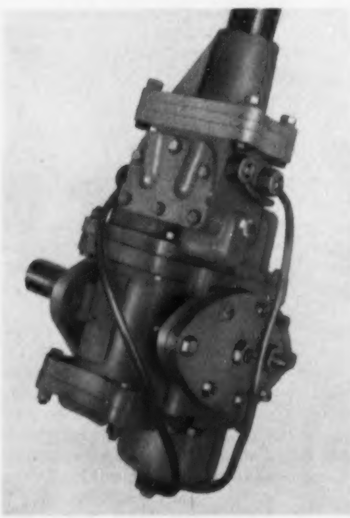


Fig. 1. As can be seen from this illustration, the Marles power assisted steering unit is a compact assembly

Fig. 2. A six-hole mounting flange is provided to suit different installations, but only four bolts are required to secure the unit

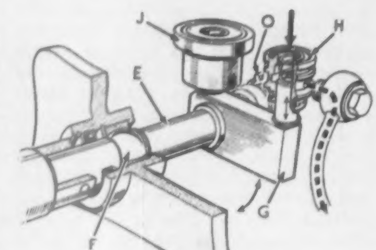
which controls the admission of oil to one of the cylinders, is some distance from the axis of the valve shaft. The axis of the other two valves, which control the escape of oil from the cylinders, is closer to that of the shaft. Each of the four valves is of the piston type; the oil is admitted axially into the centre of the valve and leaves through circumferential ports.

The distribution valves, the centres of which are permanently in communication with the pump delivery line, are always in hydraulic balance. On the other hand, the centres of the reaction valves are in communication with the appropriate cylinders, so the pressure applied to the piston component of the valve is proportional to the pressure in the power cylinder, and, therefore, to the power assistance given to the steering effort. The force due to the cylinder pressure on the reaction valve piston tends to push the valve block and shaft back into the neutral position. This can only be resisted by the torque in the steering wheel tending to displace the spur wheel sideways.

Thus, the torque at the steering wheel is proportional to the pressure applied to the reaction valve which, in turn, is proportional to the steering effort applied by the pistons. At the same time, the tangential force between the two spur gears, which causes the sideways displacement, also tends to actuate the cam and roller, and so forms the manual part of the steering effort. To alter the ratio between the torque at the steering wheel and that applied to the rocker shaft, it is necessary to use piston valves of different areas.

Operation

During operation, the initial movement of the steering wheel causes one distribution valve to close completely and the other to open fully. This cuts off the oil supply to number 1 cylinder and diverts all the output of the pump to number 2 cylinder. Then the reaction valve of number 1 cylinder is moved further from neutral towards the open position and that of number 2 starts to close. The amount of closure of this valve controls the rate of escape of oil from number 2 cylinder and, since the flow of oil is constant, the pressure.



E. Valve shaft
F. Spherical bearing
G. Valve block
H. Distribution valve
J. Reaction valve
O. Spur gear

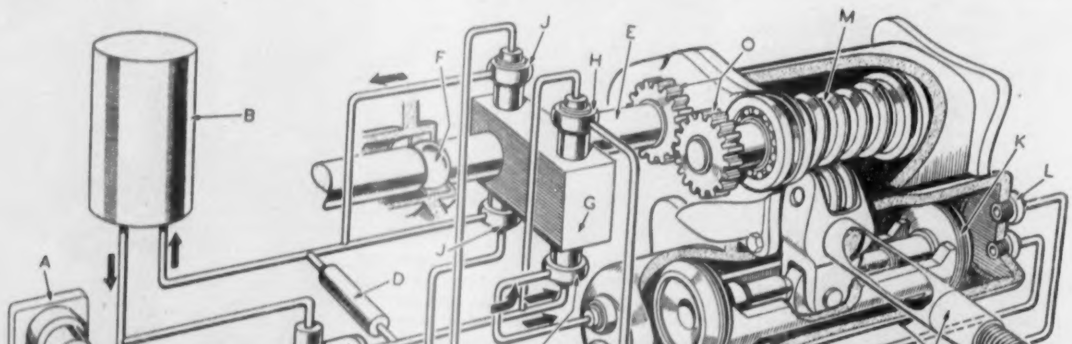
Fig. 3. Details of the valve block assembly of Fig. 4, showing only two of the valves

When the driver ceases to turn the steering wheel, the pressure continues to actuate the piston and rocker shaft until the cam spur-gear moves the valve-shaft gear back towards its neutral position. This movement opens the reaction valve slightly so that the pressure is reduced until it is no longer high enough to turn the rocker shaft. Thus, a state of balance is obtained between the steering torque on the rocker shaft, the oil pressure and the steering-wheel torque.

Movement due to an increase in torque at either the steering wheel or the rocker shaft will then cause the reaction valve to close slightly, and the pressure will be increased. Movement due to a reduction in torque has, of course, the opposite effect. It follows that any suddenly applied torque at the rocker shaft, such as may arise from a wheel traversing a rough patch on the road, will cause the valves to operate in such a direction that pressure is built up to oppose that torque. When the rate of change of the kicking force is high, the inertia of the steering wheel and column is sufficient for the valves to operate without any appreciable reaction being felt by the driver. If the rate of change of reaction is low, some small restraint must be applied to the steering wheel to operate the valves. However, the feel-back is hardly detectable until the speed is so low that there is, in fact, no sense of kick.

After completion of the turn, the driver can do one of two things. He can simply release the steering wheel, in which case, as there is no longer a restraining torque, the pressure on the reaction valve pushes the valve shaft towards the neutral position. Then all the valves open and the gear is centred again under the influence of the restoring torque from the road wheels. It is essential that there is not too much friction in the steering column bearings, otherwise the valve shaft would be prevented from centring properly and the action would be sluggish. Alternatively, the driver can positively turn the steering wheel back towards the neutral position. If the wheel is turned faster than the steering system can centre itself, the valves are displaced in the opposite direction and pressure built up to assist the driver.

A non-return valve is incorporated in the hydraulic circuit between the flow and return lines. This valve is normally held closed by the hydraulic pressure and only comes into operation



A. Pump driven by engine
B. Reservoir containing filter
C. Flow-control and pressure-relief valve
D. By-pass valve
E. Valve shaft
F. Spherical bearing
G. Valve block
H. Distribution valve
J. Reaction valve
K. Piston
L. Cylinder
M. Marles cam-and-double-roller
N. Rocker shaft
O. Spur gear

Fig. 4. Diagrammatic illustration of the Marles power assisted steering system

if the steering is operated when the pump is not running, for example, when the vehicle is being towed, or if the drive to the pump should fail. If this valve were not incorporated, there would be considerable resistance to the operation of the steering gear in these circumstances, since the oil could not be drawn through the stationary pump.

In the event of the failure of the hydraulic system, the steering gear operates as an ordinary manual type of unit, except that there is a small amount of backlash due to the valves moving the full extent of their travel before acting as stops to restrict the sideways movement of the valve shaft. The limit of their travel is 0·020 in in each direction. When it is reached, the torque from the steering wheel operates the cam in the normal manner, but through the spur gears.

A noteworthy feature of the design is that there are only three high-pressure seals on the moving components. One is on the pump spindle and two are on the pistons. The seals on the rocker shaft and the steering column are subject only to the return back-pressure. There are no seals between the moving parts in any of the valves. This avoids a possible source of friction. Leakage from the valves or past the pistons goes into the body of the unit and thus to the return line to the reservoir.

The steering unit illustrated is designed to give a rocker shaft output torque of 22,000 lb-in, with an oil pressure of 650 lb/in^2. Higher oil pressures, of up to 1,000 lb/in^2, can be adopted to give an increase in torque output. Normally, the effort at the rim of a 21 in diameter steering wheel is about 20 lb, but this can be varied by altering the diameters of the pistons in the reaction valves.

To make the unit suitable for application to many different installations, at the lowest possible cost, it has been designed so that it can be assembled in a variety of different ways. The valve box and column can be mounted in any of three different positions on either end of the main casing. In addition, the unit can be mounted with either the cam or the cylinders uppermost. This makes possible the adoption of any one of twelve alternative mounting positions. Six of these positions are for a right-hand box and six for a left-hand box. Furthermore, provision is made for either right- or left-hand operation of the gear for each position. Six bolt holes are drilled in the mounting flange, but any four are adequate for fixing the unit. Rocker shafts and nose pieces are available in alternative lengths. There is a 2 in offset between the axes of the column and the camshaft.

TRAILER BRAKING

A Clayton Dewandre and Co. Ltd. Development

HITHERTO, most braking equipment for trailer vehicles has been designed for maximum effectiveness in only one service condition—when a full payload is being carried. With such equipment, braking efficiency is low and harsh trailer braking will occur when the trailer is empty or only partially loaded. This always results in excessive tyre wear when the brakes are applied, and it may actually be dangerous, since it can result in loss of control of the trailer.

To overcome these difficulties the Clayton Dewandre Co. Ltd., Lincoln, have developed what is designated the Light Laden valve to provide adjustable braking control to suit varying trailer loadings. This device acts as a pressure reducing valve. Simple pre-setting of a handle into the appropriate one of three positions determines the maximum pressure in the brake line. The handle has a cam action which preloads a pressure spring that acts on a reaction piston to give 28 lb/in^2 for empty trailers, 57 lb/in^2 for part loadings, and full pressure for brake control under normal, full-load conditions. The reaction piston holds an inlet valve open when the brakes are off.

When the brakes are applied, the air passing through the valve to the brake cylinders acts on the piston. As the pressure rises in the brake line, to a maximum determined by the setting of the control handle, the piston reacts against the spring and allows the spring-loaded inlet valve to seat. This cuts off further air supply to the trailer brakes. When the handle is set at full pressure, the piston is positively held down so that the inlet valve remains open for maximum braking effect.

The Light Laden valve can be applied with any of the three air pressure systems for trailer brakes. With the "single line" upright system the valve is fitted behind the coupling. In a "two pipe line" system it is inserted in the service line between the coupling and the relay emergency valve. With an "inverted" system it is positioned between the control valve and the brake cylinders or chambers. The Light Laden valve may also be applied to tractor braking systems. In such an application it is fitted in the pipe leading to the rear wheel brake cylinders or chambers, and provides a varying effort at the rear axles corresponding to the load that is being carried.

CERAMICS

AN increasingly important part is being played in modern engineering by oxide ceramics which have inherent refractory properties far superior to those so far produced by metallurgists. A typical example is Hylumina, developed by K.L.G. Sparking Plugs Ltd., originally for aircraft sparking plugs that have to withstand the effects of tetraethyl lead, fluctuating temperatures and mechanical vibration. This material has since proved suitable for jet engine re-heat plugs, operating in temperatures up to 700 deg C. and for use in the thermo-couples for measuring jet pipe temperatures.

Hylumina consists of some 95 per cent fine-grained aluminium oxide, a bonding material, and very small quantities of two other ingredients, one of which acts as a flux. Too high a proportion of silica can have an adverse effect on the resistance to corrosion. In the manufacturing technique, the finely screened materials are compounded in mixers and slightly damped to give them cohesion. The mechanically mixed material is put into moulds and compressed roughly into shape. This cast is then given more exact shape with a tipped tool before it is ready for furnace treatment.

In the furnace vestibule the casts are packed into oblong boxes of fire-proof clay, which are mounted on trolleys and coupled to a slowly moving conveyor for transfer through the furnace. From the time a trolley enters the furnace tunnel until it emerges, a period of 33 hours elapses. After firing, in addition to 100 per cent visual inspection, a sample batch from each firing is given a thermal shock test up to 950 deg C and a compressive test to destruction.

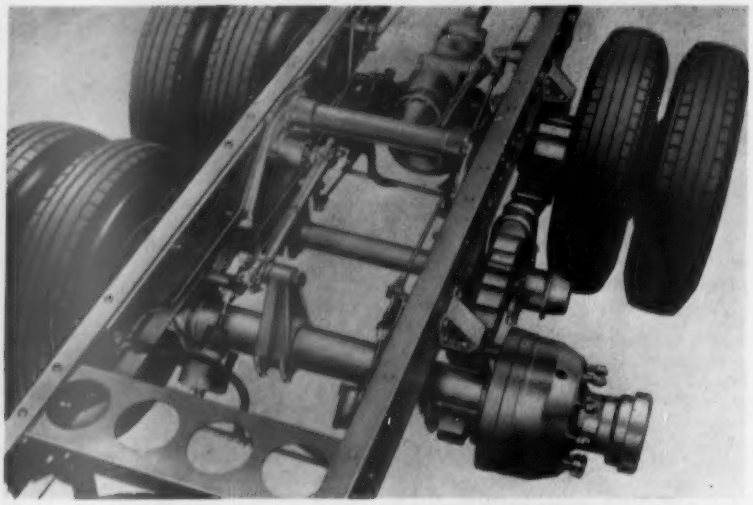
In the electronics field, K.L.G. have made an important contribution by the introduction of sealed terminals for transformers, condensers, potentiometers and other applications. In addition to the orthodox type of sealed terminal with a sealed-in conductor, a range of capped terminals has recently been developed, primarily for use where space or design limitations preclude the use of more othodox types. Further designs cater for applications where heavy currents or high voltages with an absence of corona are met and many other applications.

LEYLAND TRAILING AXLE

A New Feature that is Noteworthy for its Simplicity

IT is now nearly twenty years since Leyland Motors Ltd. last had trailing-axle equipment in their heavy duty goods range, and the recent re-introduction of this arrangement should satisfy the demands of British operators whose working conditions permit the use of this type of layout without any undue sacrifice of efficiency. The trailing axle has been introduced, for use in conjunction with a single driving axle, to extend the scope of the six-wheeled Hippo and eight-wheeled Octopus goods vehicle ranges. It is available for the home market only.

With the trailing-axle layout, the unladen weights of both models are reduced by approximately 5 cwt. This layout also makes possible a considerable price reduction. The new unit is completely interchangeable with the normal driven axle. Both the general layout and detail components such as the torque reaction tubes, air brake cylinder mountings, spring attachments, brake drums and connections, hubs, axles and tubes, are common to the driving axle and the alternative trailing axle. Therefore, the trailing axle can be substituted directly for a driven axle or *vice versa* without replacement of any other parts.

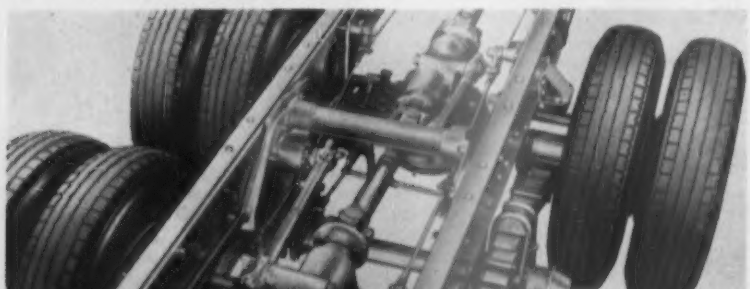


On the single-drive unit, the brake torque reaction tube is attached to a pedestal on the centre component of the trailing axle

The general layout of the rear suspension and final drive is well known and is briefly as follows. To carry the bogie, the channel section side members are reinforced locally by inserts, also of channel section. The side member and the insert are fitted together in such a manner as to form a box section. On each side of the bogie, the spring is pivoted on the ends of a cross tube. The spring eyes are pinned between lugs extending downwards from a sleeve assembled over the end of each axle. This sleeve is keyed to the axle.

Brake torque is reacted by two rods fitted with generously proportioned ball ends. One end of each rod is attached to the worm housing and the other to the attachment bracket at one end of the bogie cross tube. When a trailing axle is fitted, there is, of course, no worm housing to which to attach the torque rod, so a pedestal is mounted on the centre component of the axle to receive the end of this rod.

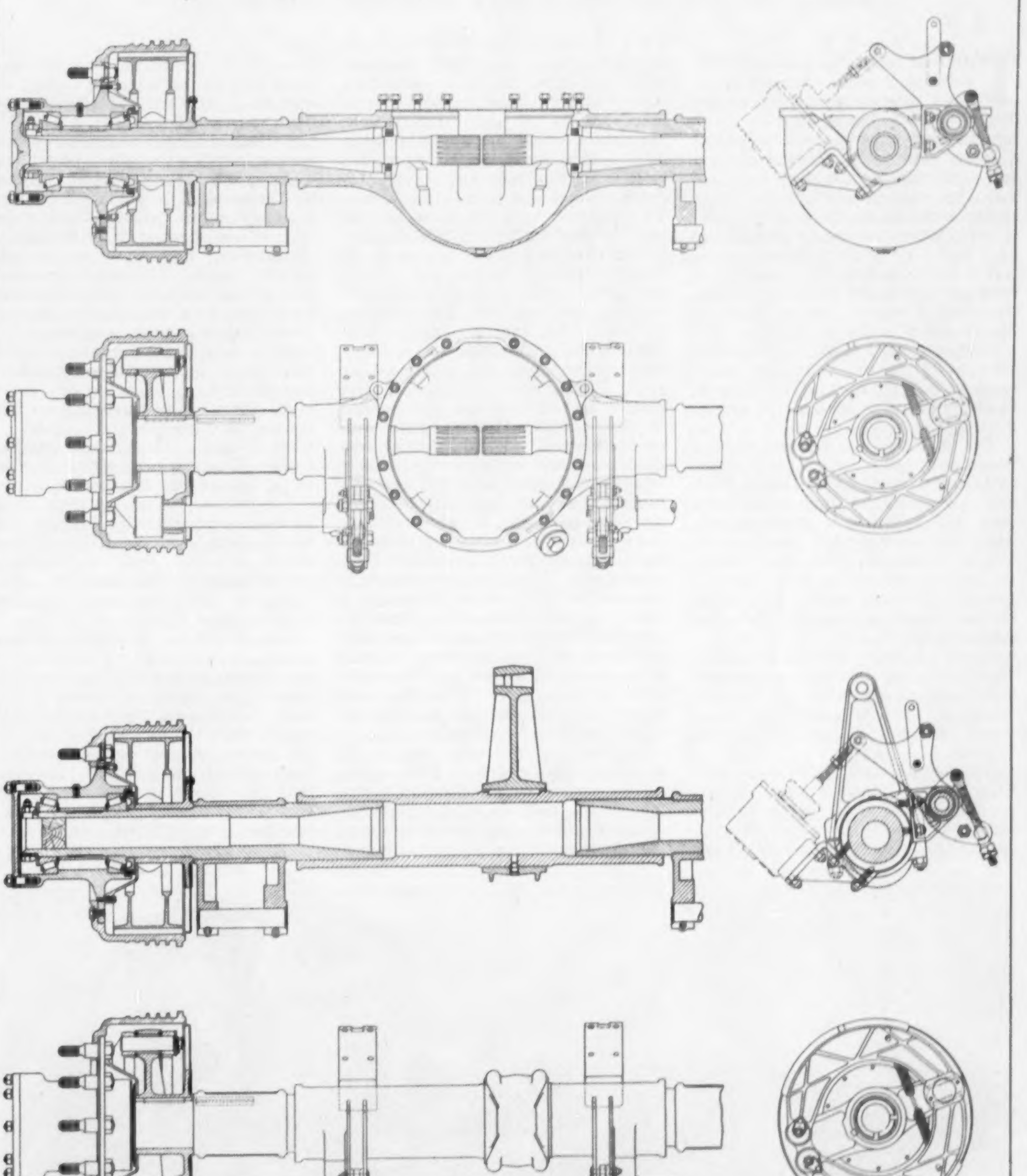
It is doubtful if a simpler layout could possibly have been devised for the trailing axle. The axle tubes, instead of being mounted in sockets on each side of the differential casing, are simply pressed into a sleeve member that takes the place of the differential unit. Thus, the whole of the assembly on each axle tube is precisely the same for both the new trailing- and the older driving-axle designs.



By comparing this illustration of the double-drive unit with that of the single-drive unit, it can be seen that the layouts are almost identical and that many components are common

Among the components that are normally mounted on the differential casing are the inner supports for the brake cross tubes. Immediately outboard of these supports are the brake actuating levers. With the trailing axle assembly, these levers and the supports are exactly the same as with the driving axle and there are mounting faces to receive them on the central tube.

The pedestal that carries the brake torque reaction rod is not unlike an engine connecting rod in appearance. It is of H-section and has at its upper end an eye to receive the pin on which is the ball end for the torque reaction rod. At the lower end of the pedestal is a much larger eye that fits round the central tube of the axle assembly. This eye is divided in the manner of an engine connecting rod big end, and the two parts are held together by four studs, which clamp the assembly to the axle tube. Location against rotation is effected by a key in the rod and a shouldered dowel in the cap.



REAR DRIVING AND TRAILING AXLE ASSEMBLIES FOR THE LEYLAND HIPPO AND OCTOPUS GOODS VEHICLE RANGES.
In the trailing axle, a simple central tube and pedestal assembly is substituted for the differential unit

NEW DIESEL TRUCK

The First Vehicle to Carry a B.M.C. Name Plate

THE first vehicle to carry a B.M.C. name plate is the seven-ton diesel truck recently announced by the British Motor Corporation. It is supplied with either right- or left-hand steering, and with forward control only, and can be supplied complete with drop-side or platform body, or chassis and cab, or chassis only. Any of these variations can be supplied in the CKD condition for export.

A B.M.C. 5·1 litre diesel engine has been adopted as the power unit. This six-cylinder engine has a bore of 95 mm and a stroke of 120 mm. The compression ratio is 16·5:1. Maximum b.h.p. is 90 at 2,400 r.p.m., and maximum torque 225 lb-ft at 1,500 r.p.m. A direct injection combustion system is employed.

The alloy cast iron cylinder block is integral with the crankcase. It incorporates detachable wet liners, fitted with rubber sealing rings at the lower end, and full length water jackets. Alloy cast iron is also used for the detachable cylinder head, which carries the valves and the rocker gear. A special alloy steel forging is used for the seven-bearing crankshaft, which is counterbalanced and fitted with a torsional vibration damper. Shell-type replaceable steel-backed, copper-lead half bearings are used. Crankshaft thrust is taken by copper-lead faced thrust washers at the front main bearing.

A forged steel camshaft, with hardened cams, is mounted at the side of the cylinder block in seven plain bearings. Timing is by triplex roller chain from crankshaft to camshaft and injection pump. A jockey tensioner with pressure oil feed maintains correct chain tension. The chain is lubricated by oil jets. Heat treated aluminium alloy, solid type pistons, with a specially designed cavity, are fitted. Each piston has three compression rings and an oil control ring. The gudgeon pins are fully floating type and are retained by steel circlips.

The overhead valves are push-rod operated through rocker gear. They are made from heat and corrosion resisting steel and the stem ends are Stellited. To ensure efficient combustion, the inlet valves are provided with masks positioned to control air swirl. The big ends of the forged steel connecting rods have the caps inclined to allow withdrawal of the piston and connecting rod upwards through the cylinder bore. Shell-type replaceable steel-backed copper-lead half bearings are fitted in the big ends, and high duty bronze bushes in the small end.

An internal gear oil pump is driven by spur gear from the front end of the crankshaft. A floating oil strainer is provided. The oil is forced through a large capacity, renewable cartridge type full flow oil cleaner into the main oil gallery in the crankcase and thence direct to the main bearings, the camshaft bearings, etc. The rocker gear is pressure fed through an external pipe from the main oil gallery.

A pneumatic governor unit, set to limit the engine speed to 2,400 r.p.m., is incorporated in the injection pump. To assist cold starting, a hand-operated excess fuel device is fitted. The mechanical fuel transfer pump is driven from the engine camshaft and feeds fuel to the injection pumps by way of a renewable cartridge type filter.

A thermostatically controlled cooling system is employed, with a centrifugal water pump belt driven in tandem with the fan. The coolant is fed into a gallery in the cylinder block, from whence it is directed on to the sleeves surrounding the injectors in the cylinder head. The water circulates through the cylinder head, the block being cooled by water passing through connecting holes in the top face.

Drive from the engine to the transmission units is taken through a Borg and Beck 12 in diameter dry single plate clutch. The gearbox gives four speeds forward (6·061 : 1, 3·473 : 1, 1·746 : 1 and 1·00 : 1) and reverse, 6·051:1. Constant mesh gears are used for all except first and reverse.

A two-piece, open tubular propeller shaft with needle-roller bearing universal joints is used. It has a large centre bearing, with the housing rubber mounted on trunnions. The two-speed Eaton rear axle has ratios of 6·14 : 1 and 8·54 : 1.

Girling 16 in diameter, internal expanding, hydraulic two-leading shoe type brakes are fitted on the front axle, with 15½ in diameter brakes at the rear. The hand brake is mechanically connected to the rear wheels only. All the brakes are fully compensated and independently adjustable. The foot brake is vacuum servo assisted. Cam Gears Ltd. high efficiency, heavy duty steering gear is fitted. It is servo power-assisted.

7-ton B.M.C. truck

STRAUSSLER-LYPSOID TRUCK

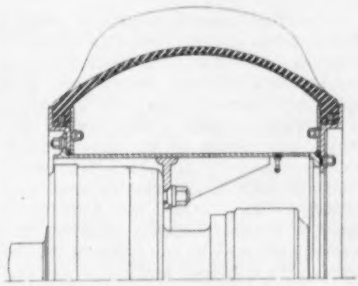
A New Cross-Country Vehicle With Two Engines and Unconventional Tyres

A NUMBER of cross-country vehicles have been designed by Nicholas Straussler and Co. Ltd., consulting engineers, of 5 Clarges Street, London, W.1. Their good performance over terrain such as sandy, swampy as well as rocky ground, is due principally to the employment of Lypsoid tyres, which are of an unusual design. These vehicles are said to have many advantages over vehicles equipped with tracks or with conventional tyres.

The tyres differ from those of conventional design in that the width between their rim beads is greater. Their radial depth is small, however, so their dimensions as measured round the periphery of their cross section are but little different from those of conventional tyres designed for the same loads. This means that the weight of the Lypsoid tyre is much the same as that of a conventional tyre with equally large treads.

The fundamental requirement for cross-country traction is the provision of tyres or tracks that give a large contact area on the ground. Because of the low ground pressures thus obtained, the vehicle can travel over sand, or soft or even swampy ground. Tracks are not an ideal solution, since a considerable proportion of the power developed by the engine is lost by friction in the track mechanism. Moreover, track life is generally relatively short, and both first cost and maintenance expenses are high. Also, for long road journeys, track vehicles must be carried on wheeled transport.

In the United States, very large diameter tyres have been employed for cross-country work. These give a reasonable area of contact, but their weight is such that mechanical handling is required to fit or remove them. The large value of the torque reaction obtained with these tyres calls for costly and heavy driving mechanisms. Furthermore, they elevate the centre of gravity of the vehicle, which tends to lead to excessive rolling when cornering. With such large diameter tyres, the rim size is restricted, so the provision of adequate brakes is not always easy.



A Lypsoid tyre assembled on a Kirkstall axle

The section of the crown of the Lypsoid tyre is an arc of large radius. Therefore, under load, the distortion of the tyre is such that the centre of the crown is bent inwards towards the rim and the shape of the area in contact with the ground is that of a shallow inverted pan. This is a good feature because it tends to compact the earth under the tyre instead of forcing it out from beneath as is the case with more conventional tyres.

Tubeless tyres of special design are employed on the Straussler vehicle to give it good cross-country performance

Another advantage of this type of tyre construction is that the angular deflection of all parts of the carcase is small. This is in contrast with conventional tyres, in which the angular deflection of the vertical walls of the carcase is relatively large. For this reason, the walls of conventional tyres have to be thin, to avoid large bending stresses, as well as strong. The reason why the angular deflection of the tyre used on the cross-country vehicle is small is that its beads are clamped to the rim. This gives an *encastré*, instead of *simply supported*, beam effect, so far as the flexure of the cross section is concerned.

As can be seen from the illustration of a Lypsoid tyre and wheel assembly on a Kirkstall axle, not only is the bead positively gripped by the rim, but also it has no inner tube. An outwardly turned flange is incorporated at the edge of the wheel periphery adjacent to the vehicle, and at the outer edge the flange is turned inwards. This is to enable the tyre to be removed without taking the wheel off the vehicle. The wheel rims are separate components; each is in two pieces and is bolted to the flange. One of the pieces is of Z-section and the other is a plain ring like a very large diameter washer which, when bolted to one flange of the Z-section, forms, in conjunction with the remainder of the section, a channel section in which is the tyre bead. Effective sealing is obtained by clamping the bead firmly between the flange of the Z-section and the ring, and also by fitting a rubber ring between this assembly and the flange round the edge of the wheel.

Tyres of this type in general are designed for air pressures of from 5-15 lb/in^2. On the vehicle that is the subject of this article, the pressure is 10 lb/in^2. Because of this low pressure, large rocks can be traversed without any daylight showing under the tyre and without appreciable vertical movement of the unsprung mass. Another advantage is that the natural frequency of the sprung mass is about 90 cycles/min and is, therefore, approximately the same as that of a normally designed car suspension. Conventional tyres, of course, have a natural frequency of 400 cycles/min or more. Vehicles fitted with Lypsoid tyres are satisfactory without conventional suspension springs and their unsprung weight is very small indeed. A tractor has been designed on this principle, but the truck has conventional semi-elliptic springs to improve its cross-country performance. This type of

spring, of course, is inexpensive and easy to maintain.

At the design stage, calculations are made to ensure that the loading and contact areas of the tyres for four-wheel-drive vehicles or for four-wheel bogies are the equivalent of a normal, well-designed track vehicle. That is, the mean ground-contact pressure ranges from slightly under 5 lb/in^2 to 10 lb/in^2. On hard metalled roads, no doubt the local pressures on the treads necessary for use on soft ground are appreciably higher than this. However, treads of different designs can be supplied to suit various purposes. The depth of the treads of the cross country truck is such that when the vehicle is steered on normal roads, practically all the distortion of the tyre takes place in the treads themselves, and their resilience is said to be such that there is no tyre-scrub while the local loading is high. On soft ground, under conditions in which ordinary tyres leave deep grooves and steering is therefore difficult, the Lypsoid tyre does not sink in, so a conventional recirculatory ball type, steering unit is adequate without power assistance. It is claimed that roll during cornering is not experienced with the Lypsoid tyres, even when the inflation pressure is as low as 3 lb/in^2. Since the weight supported per unit area of tyre contact with the ground is relatively small, it seems likely that the claims with regard to the long life of these tyres over cross-country terrain are fully justified. The rim diameter and width is such that large brake drums can be accommodated. As can be seen from the accompanying illustration, the overall width of the assembly is but little different from that of a twin-wheel arrangement.

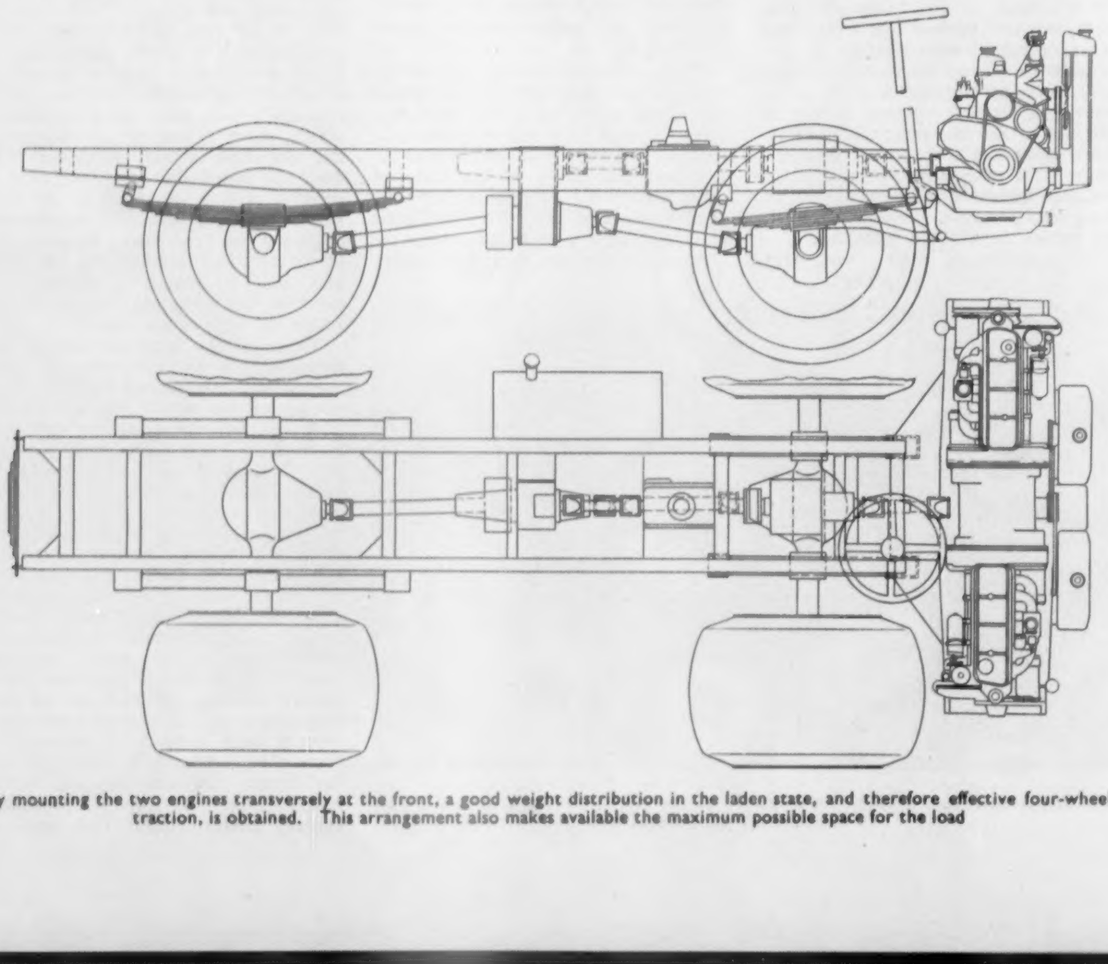
The vehicle

The cross-country vehicle is unconventional in layout. Two, four-cylinder engines are installed, back-to-back, with their crankshaft axes in line in a transverse plane. They are bolted to a common casing, which houses the bevel gear drive to the gearbox. A simple dog clutch is interposed between each engine and its bevel gear drive. Thus, when the loading is light, it is advantageous to disconnect one engine and to run entirely on the other. In this way, low fuel consumption can be obtained because instead of two engines running inefficiently at small throttle openings, one is used with a fairly wide throttle opening. Another advantage of this arrangement is that in the event of engine failure in areas remote from servicing facilities, the vehicle can still be driven home by the sound engine. Undoubtedly, this is an important feature for vehicles intended for use in many overseas countries.

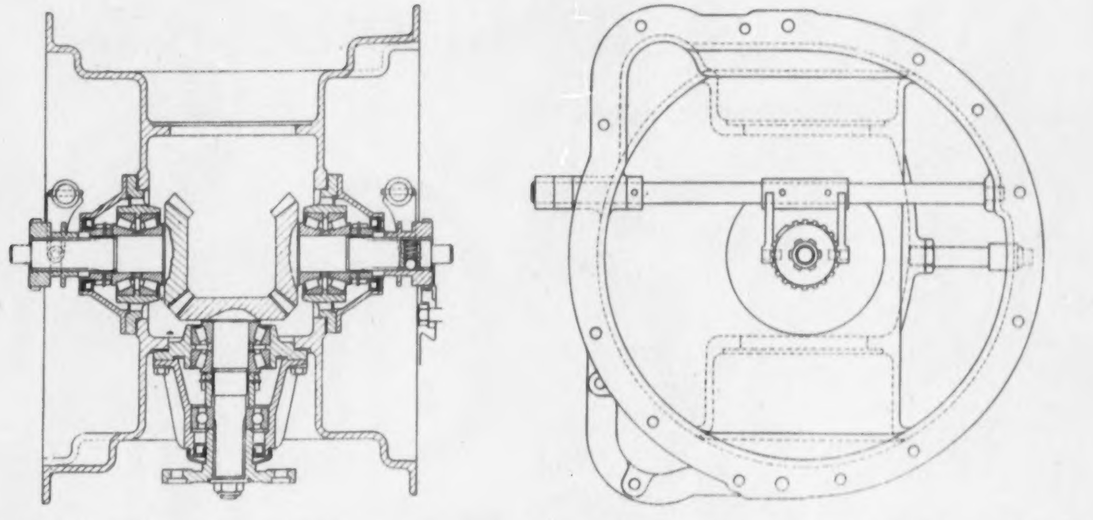
A number of different engines have been fitted to this vehicle. There is considerable advantage in employing two small units produced in large numbers. This is because they are less expensive than one larger engine that is produced in smaller quantities. Moreover, spare parts are generally readily obtainable.

The engines are mounted transversely in front of the vehicle for a number of reasons. One is that with this arrangement, the distribution of weight between the front and rear wheels, in the laden condition, is more or less equal. Therefore, the traction obtainable with four-wheel drive is better than would otherwise be possible. Another is that it enables the greatest possible area to be made available behind the driver for load carrying. Thirdly, since the frame side members are relatively close together, to allow clearance for the large angle of lock of the steered front wheels, there is not space between them to accommodate the power units. Thus, if they were installed in any other position, they would have to be higher than they are at the front of the vehicle.

The specification of the vehicle as currently designed is as follows. Two four-cylinder petrol engines are employed. Each has a bore and stroke of 100 and 115 mm respectively. The power output is 70 b.h.p. at 2,800 r.p.m. and the torque is 165 lb-ft at 1,500 r.p.m. Its compression ratio is 6:1. The two engines drive a common shaft through bevel gears, which have



By mounting the two engines transversely at the front, a good weight distribution in the laden state, and therefore effective four-wheel traction, is obtained. This arrangement also makes available the maximum possible space for the load



A simple bevel gear arrangement is used to transmit the drive from the two engines, and a dog-clutch is interposed between each engine and the bevel unit

a ratio of 23:24. A five-speed gearbox is employed; its ratios are: top 1:1, fourth 1·6:1, third 2·79:1, second 4·73:1, first 8·14:1, reverse 7·76:1. The high ratio of the two-speed transfer box is 1:1 and the low ratio 2:1. In the axle, the spiral bevel has a ratio of 7:1. This gives a maximum overall reduction of 118:1. With one engine the tractive effort obtainable is 9,000 lb and with two engines it is, of course, twice as much, that is 18,000 lb.

The tyre equipment comprises four Straussler-Lypsoid tyres, 44 in diameter × 28 in wide. In the fully laden condition, the ground pressure is only about 4½ lb/in². This is low enough for traversing even deep swampy ground. The ground clearance at the centre of the vehicle is 18 in and under the axles it is about 15¼ in. Both the approach and the departure angles are large so that the vehicle can negotiate steep banks. The turning circle is 45 ft diameter.

Unladen, the all-up weight on the front axle is 2 tons 10 cwt, and on the rear it is 1 ton 1 cwt. With a load of 2½ tons, the all-up laden weight on the front axle is 2 tons 15 cwt, and that on the rear axle 3 tons 6 cwt. The overall dimensions of the vehicle are: length 16 ft 3 in, width 7 ft 6 in, and its wheelbase is 8 ft 6 in. On soft ground the contact area of each tyre is 800 in²; this gives a total contact area of 3,200 in². The maximum speed of the vehicle on good ground is said to be 50 m.p.h.

DYSON BOILER TRANSPORTER

A Development for Oilfield Drilling Equipment

THE trailer shown in the accompanying illustrations is one of a fleet built by R. A. Dyson and Co. Ltd., of Liverpool. These units are for the transport of large, locomotive type boilers for oilfield drilling equipment, and have been built for the Assam Oil Company Ltd. The trailer frame forms a permanent mounting for the boiler, which has to be transported from site to site as drilling operations are completed. When it is necessary to move the units, pneumatic-tyred bogies are fitted; one set of bogies can handle several units in turn.

The Dyson trailer for transporting boilers for oil drilling equipment

The trailer chassis incorporates a built-in fire box and the high pressure boiler is secured to the trailer by means of turn buckles and chains. At the operating site, the two bogies are removed and the boiler is fired in position on the frame. This arrangement, whereby the bogies can be quickly and easily removed, is an important feature of the design.

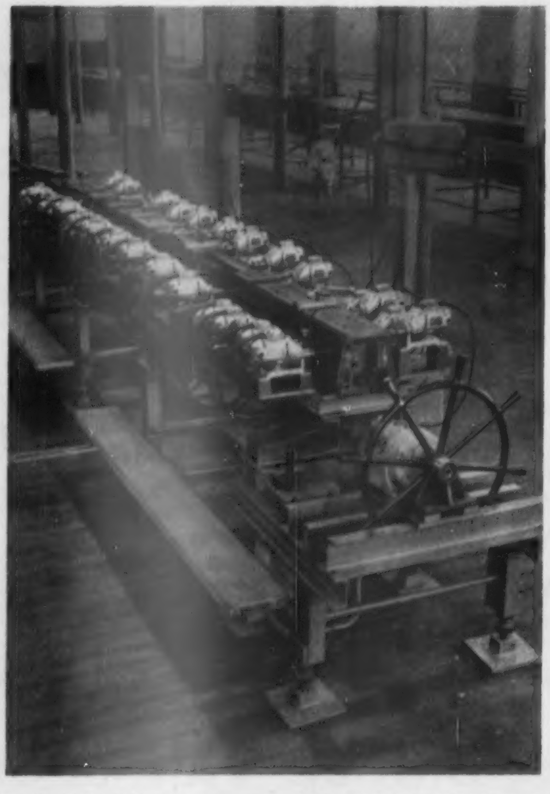
The front carriage is of the quick-release type. Operation of a lever releases the jaws that lock the king-pin, and the carriage can then be removed. This leaves the front end of the unit supported on its landing gear. The rear bogie is carried on a single cross shaft and can be released from the frame by undoing the clamp bolts. If the landing gear at the front is retracted, the boiler tips forward and the back end of the frame lifts so that the rear bogie can be rolled away. The front end of the unit can then be raised until the whole assembly is horizontal, with its base resting on the ground. It is then ready for operation.

Since each boiler weighs approximately 28 tons, the saving in time and transport costs is appreciable. Mild steel sections are used throughout for the trailer, which is 41 ft long. All the sections are electrically welded together on assembly. The rear bogie incorporates a central cross shaft on which the two balance beams can oscillate. Each of these balance beams is fitted with two stub axles. In the front carriage there are two, short, oscillating axles mounted in line on heavy-duty, trunnion-ended spring.

CHASSIS FRAME DRILLING

An Interesting Use of Black and Decker Tools

A CHASSIS frame drilling jig in which 42 Black & Decker ⅜ in. heavy duty drills are used has been put into operation by Seddon Diesel Vehicles Ltd., Oldham, Lancs. It is being used to drill all the rivet and bolt holes for the cross-members and spring hangers on the Seddon "25," a small chassis powered by a 34 b.h.p. diesel engine. Since the jig has been put into service, the time factor for chassis frame drilling has been reduced to one-third of that previously taken.



Jig for drilling chassis frames at the Works of Seddon Vehicles Ltd.

Designed by the vehicle manufacturers after consultation with Black & Decker Ltd., the jig, shown in the accompanying illustration, consists of a steel framework on which six laterally sliding carriages are arranged in pairs facing each other along the length of the jig. The portable tools are mounted on the carriages and form two banks. The design of the jig is such that two chassis frames can be placed back to back between the left- and right-hand bank of drills; when they are in position they are firmly held in place by means of screw operated clamps.

A central shaft running the length of the jig carries six pinions which mesh with teeth cut in racks attached to each carriage. Thus, if the shaft is rotated in a clockwise direction, the racks simultaneously move towards the right and *vice versa.* As the carriages are attached to the racks, they will also move in the same direction. The only modification to the power tools mounted in the jig is that the combined side handles and switches are removed, and junction boxes fitted in their place. These junction boxes are wired to master switches controlled by the operator. By this arrangement it is possible to switch on all the drills in one bank simultaneously.

To drill a pair of chassis frames, the operator switches on the left-hand bank and rotates the shaft in a clockwise direction, thus bringing the drills up to the work. If the shaft is rotated still further, the drills will enter the work and finally break through. When this occurs, the shaft is rotated anti-clockwise, thus retracting the left-hand bank and simultaneously bringing up right-hand bank, which is then switched on.

In practice, it was found one operator could not bring sufficient force to bear on the shaft to allow satisfactory operation, and a capstan wheel was therefore fitted at one end of the jig, driving the shaft through 4:1 reduction gear. This modification now enables each bank of drills to be fed into the work smoothly and without undue pressure from the operator. So that all the Black & Decker portable drills do not start drilling simultaneously, the twist drills are inserted in their respective chucks at varying positions in relation to the face of the chassis frame sides. Accuracy of the holes drilled in the frames is ensured by means of hardened steel drill bushes.

The frames when mounted in the jig for drilling are inverted, and when drilling of the side holes has been completed by the two banks of ⅜ in heavy duty drills, a number of holes are then drilled in the frame flanges. This is effected by two Black & Decker ½ in heavy duty drills mounted in No. 60 vertical stands carried on sliding arms. The position of the holes is located by means of hardened steel bushes mounted on the jig above the frames.

After all drilling operations have been carried out, the frames are removed and all holes deburred by means of a countersink mounted in a Black & Decker ¼ in heavy duty drill. When this operation has been completed, the frames are bent to shape and passed on to the assembly line. Should any modifications be made to the chassis frame, the rig can easily be modified to suit.

PORTABLE POWER HACK SAW

THE Hand-I-Hack portable power hacksaw developed by Lipe-Rollway Corporation of America, is a tool with many uses for almost every engineering factory. Gaston E. Marbaix, Ltd., Devonshire House, Vicarage Crescent, London, S.W.11, are the British agents for this tool, which is designed to eliminate the need for lengthy and tiring hack-sawing by hand.

Although the tool is rugged, it is light in weight. It can be attached to the work by means of an integral swivel-type vice, and will support itself in any position, horizontal, vertical or angular. The sawing motion is draw cut and lift return. It is effected by an ingenious patented motion, which is fully adjustable from the pressure required for cutting 3 in mild steel to a feather light touch for cutting thin wall tubing.

The vice has a sliding jaw that is grooved to take round stock, and a stationary screw with a hinged lever. It is adjustable and is accurately calibrated for cutting angles up to 45 deg. Power is supplied through a ¼ h.p. motor, which can be plugged into any convenient circuit.

THE RING-SPRING

A Type of Spring Suitable for Trailer Couplings

In this country, the ring-spring is relatively little known, although it is used extensively on the Continent for railway buffers and couplings and, more recently, for aircraft landing gear, and lorry and bus trailer couplings. Because the properties of the ring-spring are not well known here, attempts are sometimes made to use it in applications for which it is not suited: for instance, it has been suggested as being particularly attractive in connection with vehicle suspensions, an application for which, in fact, it has scarcely any merit.

The ring-spring, an invention of Dr. Ing. H. C. Ernst Kreissig, M.I.Loco.E., originated from an appreciation of the fact that springs stressed either wholly in compression or wholly in tension are, in general, uniformly stressed over their entire length and cross section and, so far as efficiency of material utilization is concerned, are unsurpassed by any other type of spring, but the space requirements of such purely compressive or tensile springs, for example, a wire rope, are very unfavourable. It was this aspect of the design that led to the development of the ring-spring.[1]

This spring consists of a number of rings with conical seating faces assembled as shown in Fig. 1. Axial load applied to such an assembly is transformed, by the wedge action of the rings, into substantially larger radial forces, which elastically expand the outer rings and compress the inner ones. As a result of these alterations in diameter, the length of the column is reduced in proportion to the load. The circumferential stresses in the rings are approximately uniform; this gives a high efficiency, on the basis of energy storage per pound of metal. Because of the friction between the conical surfaces, a much higher spring constant is obtained for the compression stroke than for the return stroke, the spring absorbing only about two-thirds of the energy dealt with in compression, Fig. 2.

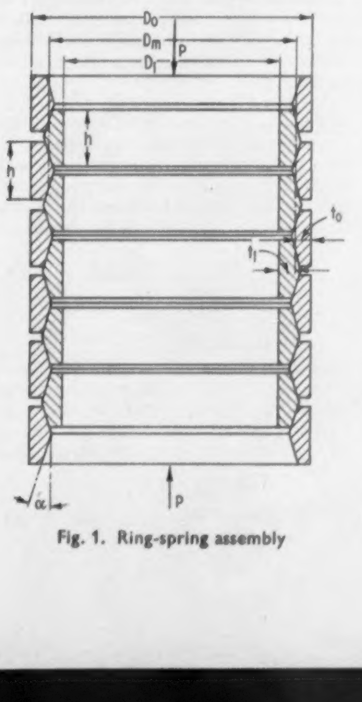


Fig. 1. Ring-spring assembly

A ring-spring that was originally developed for heavy wagon buffers was designed to deal with a compression force of 30 *t*, and the rebound force was 12·7 *t*. The energy absorption was 9,500 ft-lb. Other assemblies capable of dealing with 45 and 60 *t* and absorbing 11,000 and 11,500 lb-ft respectively, were also produced. The 45 *t* assembly was applied to heavy goods wagons, carriages and locomotives, its use being of advantage not only when shunting, etc., but also in reducing surge force set up in long goods trains equipped with continuous brakes.[2]

The rings are manufactured by rolling, and to facilitate both rolling and heat treatment, the outer periphery of each outer ring is slightly concave in cross section, while the inner periphery of each inner ring is convex. If seizure is to be prevented, the cone angle must be larger than the friction angle characteristic for the material. The friction angle amounts to approximately 9 deg for unmachined heavy rings, 8 deg 30 sec for unmachined light rings and about 7 deg for light, machined rings. For safety, the value of the ring cone angle α is based on the value of $\tan\alpha = 0{\cdot}25$ to $0{\cdot}3$, that is, $\alpha = 14$ deg 3 min to 16 deg 42 min, although with machined, light rings it can be reduced to 12 deg.

Ring height *h*, Fig. 1, must not be unduly small; otherwise, because of insufficient guidance, jamming may occur. On the other hand, the ring height must not be unduly great in relation to its thickness, since this may lead to difficulties due to rapid cooling during the rolling process. It is advisable to maintain a ratio of $D_o/h =$ 5 : 1 to 6 : 1. The maximum tensile stresses in the outer ring normally should not exceed 140,000 lb/in², while the permissible compressive stresses in the inner ring may be as high as 170,000 lb/in², although higher values are acceptable for components such as buffer springs, which are loaded only at relatively infrequent intervals. With these it is necessary to lubricate the rings with gear type grease, and the provision of a circumferential groove on the outer surface of each inner ring has proved to be of advantage.

The main design parameters can be determined on the basis of the ability to do work, or the elastic energy storage capacity S_e, where:

$$S_e = (V_t\sigma_t^2 + V_c\sigma_c^2)/E$$

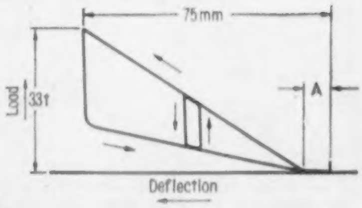


Fig. 2. Characteristics of a ring-spring designed for a 33 t load. The horizontal portion A is due to the inclusion of two slotted rings

and V_t = volume of outer (tension) ring
V_c = volume of inner (compression) ring
σ_t = maximum tensile stress in outer ring
σ_c = maximum compressive stress in inner ring
E = modulus of elasticity

Since with dynamically stressed springs, the compressive stresses are appreciably higher than the tensile ones, it is necessary to introduce a correction factor *k* so that:

$$\sigma_c = k\sigma_t$$

The value of *k* depends upon application, ring thickness, surface finish and spring steel hardness. For buffer springs the value of *k* is about 1·25, while for higher frequency applications $k = 1{\cdot}45$ to $1{\cdot}65$.

The calculations are complicated by the necessity for taking into consideration different values of maximum stresses for the inner and outer rings, but the general dimensions can be readily determined by assuming, as a first approximation, identical stress values for both rings. As an example of the method of size determination, a heavy coupling spring will be considered as follows. The load home is $P = 30\ t$, the deflection 3 in and *A*, the energy absorbed, = 7,300 ft-lb. The outside diameter is limited to 6·5 in and the height home to 16 in. A value of 1·25 is assumed for *k*, $\tan\alpha = 0{\cdot}25$ and μ, the coefficient of friction between the ring faces, = 0·16. Thus the energy absorbed by elastic deformation of the ring is:

$$A_e = A\left[\frac{\tan\alpha\,(1-\mu\tan\alpha)}{\tan\alpha+\mu}\right]$$

$$= \frac{7{,}300\times 12\,(1-0{\cdot}16\times 0{\cdot}25)\times 0{\cdot}25}{0{\cdot}25+0{\cdot}16}$$

$$= 51{,}200 \text{ in-lb.}$$

For σ = 140,000 lb/in², and E = 30,000,000 lb/in², the volume is:

$$V = 2AE/\sigma^2$$
$$= 2\times 51{,}200\times 30{,}000{,}000/140{,}000^2$$
$$= 156{\cdot}5 \text{ in}^3$$

With D_o and D_i as the outer and inner diameters, respectively:

$$V = \pi/4\,(D_o^2 - D_i^2)\times 16$$
$$= 156{\cdot}5 \text{ in}^3$$

so that $D_o^2 - D_i^2 = 12{\cdot}5$, $D_i^2 = 6{\cdot}5^2 - 12{\cdot}5 = 29{\cdot}7$ and $D_i = 5{\cdot}45$ in. Since

both rings are assumed to be subject to the same maximum stress, 140,000 lb/in², the mean ring thickness t should also be identical, that is, $t_o = t_i$. However, as mentioned before, the inner rings can be stressed to a higher value of $\sigma_c = k\sigma_t = 1{\cdot}25 \times 140{,}000 = 175{,}000$ lb/in². For this spring, the mean diameter is:

$$D_m = (D_o + D_i)/2 = (6{\cdot}5 + 5{\cdot}45)/2 = 5{\cdot}975 \text{ in}$$

and

$$t_o = t_i = (D_o - D_m)/2 = (6{\cdot}5 - 5{\cdot}975)/2 = 0{\cdot}2513 \text{ in}$$

If the inner ring is to be stressed up to the value of $\sigma_c = k\sigma_t$, then the mean radial thickness t'_i will be smaller than the above value of t_i; consequently:

$$t'_i = 0{\cdot}2513 \times 1/1{\cdot}25 \approx 0{\cdot}2 \text{ in}$$

while the inner diameter of the new spring will be:

$$D'_i = D_m - 2t'_i = 5{\cdot}975 - 0{\cdot}4 = 5{\cdot}575 \text{ in}$$

Similarly, the volume of all the smaller diameter rings is reduced from $V/2$ to $V/2k$; this also affects the capacity of the assembly to do work. The magnitude of this effect can be determined from:

$$A_e' = \frac{V\sigma_t^2}{2 \times 2E} + \frac{V}{2k} \quad \frac{\sigma_t k^2}{2E} = \frac{V\sigma_t^2}{2E} \quad \frac{k+1}{2} = A_e \frac{k+1}{2}$$

To ensure that the original value of A_e is retained, it is necessary to divide $t_o + t'_i$ by $(k+1)/2$ to obtain the requisite value of mean ring thickness. This alters t_o to t_o', and t'_i to t''_i, respectively. Thus:

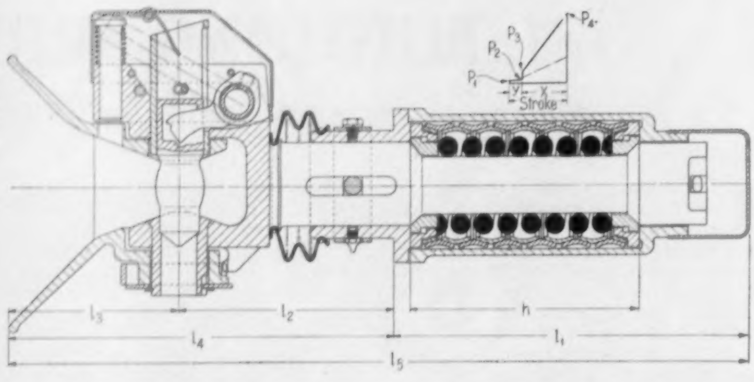
$$t'_o + t''_i = (t_o + t'_i)/\ [(k+1)/2] = 0{\cdot}4513/1{\cdot}125 \approx 0{\cdot}4 \text{ in}$$

$$t''_i = 0{\cdot}4/2{\cdot}25 = 0{\cdot}1775 \text{ in}$$

and $t'_o = 0{\cdot}4 - 0{\cdot}1775 = 0{\cdot}2225$ in

Consequently, the data for the spring will be:

$$D_o = 6{\cdot}5 \text{ in}$$
$$D'_m = 6{\cdot}5 - 2 \times 0{\cdot}2225 = 6{\cdot}055 \text{ in}$$
$$D''_i = 6{\cdot}5 - 2 \times 0{\cdot}4 = 5{\cdot}7 \text{ in}$$



Total weight of trailer, t	mm								lb				Work done ft-lb	Weight lb
	l_1	l_2	l_3	l_4	l_5	h	x	y	P_1	P_2	P_3	P_4		
3 to 6	250	165	135	300	550	161	19	4	835	1,000	3,200	1,950	500	60
6 to 16·5	285	180	135	315	600	183	23	5	950	1,400	5,150	29,300	1,040	75
Over 16·5	285	180	135	315	600	183	31	5	1,200	1,640	5,350	41,000	2,000	92·5

Fig. 3. Truck trailer assembly

$$V'_i = \pi/4\,(6{\cdot}5^2 - 6{\cdot}055^2) \times 16 = 69 \text{ in}^3$$

$$V = \pi/4\,(6{\cdot}055^2 - 5{\cdot}7^2) \times 16 = 53{\cdot}5 \text{ in}^3$$

and the energy absorbed by elastic deformation will be:

$$A_e'' = \frac{69 \times 140{,}000^2 + 53{\cdot}5 \times 175{,}000^2}{2 \times 30{,}000{,}000} = 50{,}000 \text{ in-lb}$$

The centroid diameter of the outer ring is obtained as follows:

$$D_{oc} = (D_o + D'_m)/2 = (6{\cdot}5 + 6{\cdot}055)/2 = 6{\cdot}277 \text{ in}$$

Similarly, for the inner ring:

$$D_{ic} = (D'_m + D''_i)/2 = (6{\cdot}055 + 5{\cdot}7)/2 = 5{\cdot}878 \text{ in}$$

The increase in outer ring diameter due to tensile stress will be:

$$\lambda_o = D_{oc} \times \sigma_t/E = 6{\cdot}277 \times 140{,}000/30{,}000{,}000 = 0{\cdot}0292 \text{ in}$$

while the inner ring diameter will be reduced by:

$$\lambda_i = D_{ic} \times \sigma_c/E = 5{\cdot}878 \times 175{,}000/30{,}000{,}000 = 0{\cdot}0342 \text{ in}$$

The axial deflection for each pair of conical surfaces, that is, for an assembly consisting of half an outer and half an inner ring, is given by:

$$f = (\lambda_o + \lambda_i)/2 \tan \alpha = (0{\cdot}0292 + 0{\cdot}0342)/2 \times 0{\cdot}25 = 0{\cdot}1268 \text{ in}$$

For a total axial deflection of 3 in, the number of cone surfaces or elements required is:

$$n = 3/0{\cdot}1268 = 24$$

while the height of each ring must be:

$$h = 2l/n = 2 \times 16/24 = 1{\cdot}335 \text{ in}$$

The total height of the complete spring will be the height home, plus the axial deflection, plus a safety margin of about 0·75 in to accommodate occasional overloads. Thus:

$$H = 16 + 3 + 0{\cdot}75 = 19{\cdot}75 \text{ in}$$

As well as the stresses already dealt with, additional stresses are produced by the alteration of ring diameter consequent upon load application. If a straight bar, that is, a bar with a curvature radius $r = \infty$, is bent so that the centroids of the section lie on a radius r_o, the bending moment is:

$$M = I\,E/r_o$$

where I is the moment of inertia of the section, and E is Young's Modulus. If the rod is bent again to conform with a radius $r_o + \lambda_o/2$, the required moment will be M', and to increase the radius from r_o to $r_o + \lambda/2$ will call for a moment:

$$M'_o = I\,E\left(\frac{1}{r_o} - \frac{1}{r_o + \lambda_o/2}\right) = \frac{I\,E\lambda_o}{2r_o\,(r_o + \lambda_o/2)} \approx I\,E\lambda_o/2r_o^2$$

Since $r_o = D_{oc}/2$, $\lambda_o = D_{oc}\sigma_t/E$ and $M_o = I\,E\sigma_b/e$, where σ_b is the bending stress, and e, in, the distance of the most highly stressed fibre from the neutral axis. The value of σ_b can be obtained from:

$$I\,\sigma_b/e = (D_{oc}\sigma_t/E\ 2r_o^2)\ I\,E$$

For the outer ring

$$\sigma_b = (e/r_o)\,\sigma_t$$

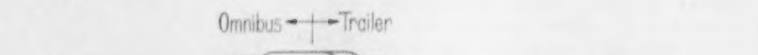


Fig. 4. Omnibus trailer assembly

while for the inner ring

$$\sigma_b' = (e'/r_i)\sigma_c$$

If $e = 0{\cdot}175$ in,

$$\sigma_b = 0{\cdot}175 \times 140{,}000 \times 2/6{\cdot}277$$
$$= 7{,}800 \text{ lb/in}^2$$

and if $e' = 0{\cdot}15$ in

$$\sigma_b' = 0{\cdot}15 \times 175{,}000 \times 2/5{\cdot}878$$
$$= 8{,}950 \text{ lb/in}^2$$

The axial projection of the ring surface is:

$$F = \pi D'_m\,(h/2) \tan \alpha$$
$$= \pi \times 6{\cdot}055 \times (1{\cdot}335/2) \times 0{\cdot}25$$
$$= 3{\cdot}17 \text{ in}^2$$

so that the maximum surface pressure is:

$$p = 30 \times 2{,}240/3{\cdot}17$$
$$= 21{,}200 \text{ lb/in}^2$$

Of this, 30 per cent gives rise to a tensile stress at the cone surface. Thus, the maximum stress in the outer ring will amount to:

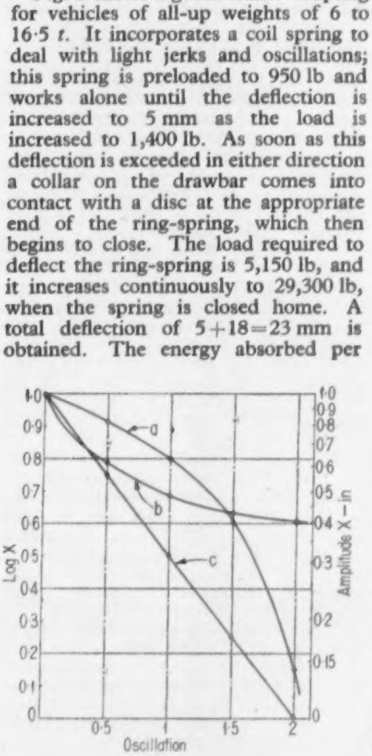
$$\sigma'_t = 140{,}000 + 7{,}800 + 0{\cdot}3 \times 21{,}200$$
$$= 154{,}160 \text{ lb/in}^2$$

while for the inner ring

$$\sigma'_c = 175{,}000 + 8{,}950 - 0{\cdot}3 \times 21{,}200$$
$$= 177{,}600 \text{ lb/in}^2$$

In the case of springs subjected to frequent loading, the maximum tensile stress imposed upon the outer rings should be below the endurance limit, but the inner rings can be stressed in compression right up to the yield point. The energy-absorbing properties of the ring-spring make it eminently suitable for use with trailer couplings; two typical designs are illustrated in Figs. 3 and 4.

Fig. 3 shows a goods trailer coupling for vehicles of all-up weights of 6 to 16·5 *t*. It incorporates a coil spring to deal with light jerks and oscillations; this spring is preloaded to 950 lb and works alone until the deflection is increased to 5 mm as the load is increased to 1,400 lb. As soon as this deflection is exceeded in either direction a collar on the drawbar comes into contact with a disc at the appropriate end of the ring-spring, which then begins to close. The load required to deflect the ring-spring is 5,150 lb, and it increases continuously to 29,300 lb, when the spring is closed home. A total deflection of 5+18=23 mm is obtained. The energy absorbed per stroke of the ring-spring amounts to about 1,000 ft-lb.



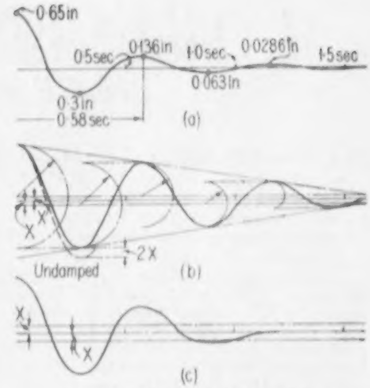
a Friction damping
b Hydraulic damping
c Viscous damping

Fig. 5. Effect of damper type on vibration decay

The coupling shown in Fig. 4 is designed for use with omnibus trailers, and incorporates a spring that is similar to that in the unit illustrated in Fig. 3. However, to avoid horizontal oscillations that might be unpleasant for the passengers, the helical spring has been eliminated; in addition, the coupling has been designed to eliminate horizontal play so far as possible. The portion of the coupling attached to the towing vehicle incorporates a grease lubricated ball joint and weighs 62 lb, while the end attached to the trailer weighs 55 lb. By itself, the ring-spring assembly weighs only about 6·5 lb.

Ring-springs also have been applied successfully to aircraft landing gear. In this application, considerable energy has to be absorbed as the result of landing impact. They have also been applied from time to time, to vehicle suspensions, both road and rail, mainly because of their damping characteristics, which are represented by the hysteresis area of the load deflection diagram. However, this diagram shows equally clearly that the ring-spring is scarcely suitable for vehicle suspensions, since the return stroke of the system can only begin when the load is reduced to about one-third of that required to compress the spring. If the load is reduced to this level, the spring will begin to expand, and if the load is then increased again, expansion will cease and the spring remain rigid until the load has been increased to about three times that at which expansion stopped; only then can compression start again. This characteristic, which is clearly shown by the small loop inside the hysteresis loop of Fig. 2, makes the ring-spring very harsh for vehicle suspensions.

Fig. 5 shows the effect of damping on oscillation decay, with reference to the fundamental differences between friction, hydraulic and viscous damping. With friction damping the damping force is inversely proportional to the square- or cube-root of velocity; with hydraulic damping it is proportional to the square of the velocity; and with viscous damping it is directly proportional to velocity. In Fig. 6, the effects of viscous and friction damping are illustrated for a four-wheeled vehicle. Each of the four springs of this vehicle have a rate, *c*, of 2,200 lb/in, the total sprung weight being 13 *t*. For a damping factor of $D = 0{\cdot}25$ of the critical or aperiodic damping, the damping constant is 102 lb-sec/in. The oscillations due to a dynamic deflection of 0·65 in will be damped in about 1·5 sec, the datum line having been crossed four times. For friction dampers, the ratio *x* between consecutive amplitudes is constant, that is, $x = k/c$ where *k* is the damping force. If $k = 200$ lb, then $x = 200/2{,}200 = 0{\cdot}091$ in; the resultant damping, as well as that for $k = 400$ lb, is also shown in Fig. 6. It can be seen that low force friction damping is of some use with small amplitude high frequency vibrations, but to be effective with oscillations



(a) Viscous damper, p=102 lb-sec/in
(b) Friction damper, k=200 lb
(c) Friction damper, k=400 lb

Fig. 6. Effects of viscous and friction damping

of the type set up by passage over rough roads or across country, ring-springs would have to be so harsh as possibly to upset the normal working of the suspension system, since forces less than those required to overcome the damper force will be transmitted in full.

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PLASTICS PATTERNS

EPOXY resins are now being produced for making plastics patterns. Apparently, they offer marked advantages over other resins that have been tested for this purpose. Aero Research Ltd., who are responsible for the development of Araldite Casting Resin M for pattern making, state that shrinkage on setting is very low indeed and is negligible when fillers are used. It is also claimed that the hardened resin is tough, durable and dimensionally stable. Araldite Casting Resin M sets cold in 24 hours, or in a shorter time if gentle heat is applied.

Since shrinkage and brittleness have been regarded hitherto as inherent defects of plastics patterns, the introduction of the epoxies into this field undoubtedly will arouse considerable interest. It should contribute significantly towards reducing costs. Full details of this resin are obtainable from Aero Research Ltd., Duxford, Cambridge.

RECENT PUBLICATIONS

Brief Reviews of Current Technical Books

The Grand Prix Car, Vol. II.

By Laurence Pomeroy, F.R.S.A., M.S.A.E.

London: MOTOR RACING PUBLICATIONS LTD., 13, Conway Street, Fitzroy Square, W.1. 1955. 8½ × 11. 344 pp. Price 75s.

This, the second volume of the second edition of "The Grand Prix Car," begins where the first volume ended: it tells the story of racing from 1939 to 1953, and gives some glances at the technical developments of 1954. The contents fall into four main groups. First, two chapters summarize the whole of post-war racing and give the results, including the fastest lap, for all the major events held under Formula I from 1947 to 1951, and under Formula II from 1952 to 1953. The next five chapters are devoted to detailed descriptions of Formula I and Formula II cars. They deal not only with types such as the 1½ litre Alfa Romeo, the B.R.M., the Largo Talbot, the 4½ litre and 2½ litre Maserati, but also vehicles such as the Connaught, Cooper, Arsenal C.T.A., the Porsche-designed Cisitalia and the twin-camshaft, 1½ litre Ferrari

A year-to-year analysis of how average lap speeds have varied in the past fifty years or more is also given. In it, the author shows that the fastest road racing car the world has yet seen is the 3 litre Mercedes Benz, the speed of which was closely rivalled by the post-war Alfa Romeo. The basis of comparison of this review is the 1906 Renault, winner of the Grand Prix in that year, which, it is estimated, would lap the Silverstone circuit at 61·8 m.p.h. Following these first chapters are fourteen more showing how racing cars developed prior to 1939 and also the reasons for these developments. Comments are given on the influence of front wheel brakes, as well as on the use of alcohol fuel, the design problems of independent suspension, general constructional methods and supercharging.

In the last two chapters, the author surveys the whole field of racing car development to the end of 1954. Among the features given in a postscript are the economics of Grand Prix racing, the relation between type of driver and type of car, and the effect of driving method and team tactics. In appendices, summaries are given, mainly in tabular form, of developments between 1900 and 1953, the specifications of 72 leading racing cars built in this period, and a list of Grand Prix cars with the number of first places they have gained in major events in Europe.

Operational Qualities of our Automobiles (Ekspluatazionnie Kachestva Otechestvennih Avtomobilei)

By D. P. Velikanov, Dr. Sc. tech.

MOSCOW: STATE SC. TECH. PUB. OF MACHINE CONSTR. LITERATURE, 1953. 5¾ × 8¾. 166 pp. 63 illus. Price 6 Roub. 15 kop. (5s. 6d.).

This book was written as a result of the directives issued during the 19th Convention of the Communist Party dealing with the last (1950-1955) five year plan when it was decided to increase road transport capacity by 80 to 85 per cent, and at the same time improve vehicle utilization and reduce operating expenses.

The first chapter deals with the overall effectiveness, operational expenses, the main points to be considered as far as comfort and operational simplicity are concerned, safety and ability to negotiate poor roads. It concludes with a five-page table summarizing the various operational requirements and the influence of the various factors upon them. The next chapter considers the load carrying capacity of six lorry types. Following this, the author considers the space available in cars and buses in considerable detail against the background of minimum requirements stipulated by the Ministries concerned. Next, the author deals with the power plant and vehicle speed data, giving the power output and fuel consumption curves of seven car engines. He also gives vehicle tractive effort curves, tables of main engine data, and performance data for four cars and nine lorries, their maximum speeds, acceleration, brake performance, and mean operational speeds under a wide variety of conditions in towns and over country roads, with and without trailers, are given. Following this, the fuel consumption is considered in some detail with the help of graphs showing the consumption in 1/100 km as a function of vehicle speed, and here the effects of load, speed, road traffic and weather conditions, etc., are clearly brought out. In addition, details are given of "norm" fuel consumption and best consumption values.

The fifth chapter deals with the life of components and as might be expected, particular attention is given to piston and cylinder wear of such cars as Moskvich and Pobeda SgS 101, 110 and 150 as well as gAS51 and gAS200 and 204. Comprehensive data are also given on the mileage between main overhauls which for good roads varies between 66,000 and 200,000 km for cars, as compared with 66,000 and 104,500 km for lorries and 132,000 and 187,000 km for buses.

Two chapters deal with experience concerning component strength. They give details and incidence of typical failures, together with the steps taken to avoid them. In addition, ground clearance, climbing angles, ground pressures front and rear wheel track and the influence of differences in track width on vehicle performance over soft ground, the effect of coefficient of adhesion on the climbing angle and useful load of typical vehicles and tractive effort data of cars and lorries are also given.

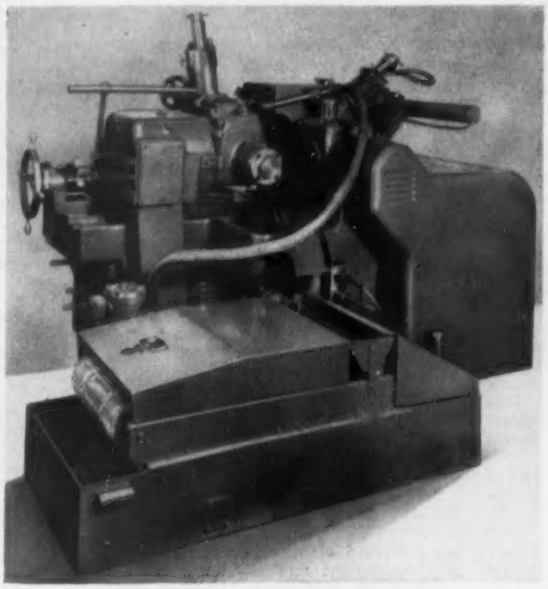
Stability is considered next and once again the general considerations are augmented by detailed data on most Russian vehicles. The next chapter deals with riding qualities and includes particulars of spring and tyre rates, ratios of spring to unsprung weights and the k^2/ab ratios for a number of vehicles together with the results of some vibration damping tests. Lorries are considered as well, though, in this, "comfort" considerations are replaced by considerations relating to the ease of loading and unloading and the prevention of freight damage in transit as well as protection from dust and rain. Ease of control is stressed by the number of gear changes required in each gear per 100 km for a number of cars and lorries operating under different road and traffic conditions and the drivers' seats are also considered. The available instruments and their accuracy are criticized and some details are given of the length and width of the "blind zones" faced by the drivers of cars, lorries and buses.

At the end, particular reference is made to the weak points in the design of more recent cars such as Moskvich, Pobeda, gAS-51 and SGM cars and gAS-51, SgS-150 and gAS-200 lorries. The book is of considerable interest not only because of detailed information on the characteristics and performance data of a wide range of Soviet vehicles, but also because it deals with their weaknesses and makes suggestions for their rectification. The data presented by the author are largely based on the results of State organized trials of new vehicles which have to cover large distances over widely varying road conditions throughout the year and it is of interest to learn from the author's recommendation that, in spite of the importance attached to the development of road transport, Russia does not possess a suitable test track of the type developed by M.I.R.A. or F.V.R.D.E.

PHILIPS UNIVERSAL CLARIFIER

Equipment for the Automatic Removal of Swarf and Abrasive from Grinding Coolant

IN all wet grinding operations it is desirable that the coolant be maintained in clean condition. Should it be allowed to become even lightly loaded with metallic swarf and with abrasive from the grinding wheel the finish of the work will inevitably suffer. Magnetic filters, if well located in the coolant circulating system, can hold most of the metal swarf, but not, of course, the abrasive. The benefits conferred by large settling tanks are somewhat illusory since settling only occurs when the machine is idle and the coolant flow is stopped. Whilst the coolant is in circulation fine particles of swarf and abrasive are held constantly in suspension and continue to exercise a deleterious effect on the quality of the work. Settling tanks must frequently be emptied and cleaned. To avoid interruption of production, this dirty operation must be performed in idle time unless duplicate tanks can be provided.



Arrangement of Type 7733/23 clarifier on Cincinnati Filmatic No. 2 centreless grinder

The Universal Clarifier, manufactured by the Industrial Division of Philips Electrical Ltd., Century House, Shaftesbury Avenue, London, W.C.2, offers a solution to the problem by trapping all types of solid matter in the coolant, including the fine suspended particles, by means of a special filter medium. The equipment continuously filters the coolant whilst the machine is working, the medium being advanced automatically as required or at predetermined intervals in co-ordination with the machining cycle.

In operation the contaminated coolant leaving the machine is fed into a metal trough, from the base of which it is delivered on to the filter medium below. Drawn from a roll mounted at one end of the apparatus, the medium is supported on the upper run of an endless wire mesh conveyor. Being loosely tensioned, the wire mesh assumes a trough-like shape under the weight of the coolant. Though the filter medium retains all the swarf, it is sufficiently porous to permit a free and uninterrupted flow of filtered coolant to the sump of the apparatus, whence it is returned by a pump to the grinding head.

As the filter medium gradually becomes choked with swarf the level of the coolant in the filter bed rises and lifts a float which, at a pre-set height, operates a micro switch to start the conveyor drive motor. The wire mesh conveyor belt moves forward, bringing new filter medium into position and automatically depositing the loaded paper into an open sludge bin. Commonly, a movement of from 2 to 4 in is adequate.

Rated at ¼ h.p., the motor and the contactor are mounted on the side of the coolant tank and fully protected against accidental flooding. All the working mechanism of the clarifier is assembled on a chassis and detachable as a unit from the tank. The motor-driven coolant pump is housed in a small compartment, which may be located at either side of the tank to give the more convenient layout. Since the tank provides ample coolant storage capacity, it is generally possible to dispense with the normal settling tank. In such circumstances no additional floor space will be occupied.

This clarifier operating in conjunction with a Lumsden 71 LEOD surface grinder separates out 68-80 lb of cast iron swarf per day

Four standard models are produced, having approximate flow rates for soluble oil coolants of 500, 1,000, 1,800 and 3,000 gal/hr. For these models the standard filter medium is supplied in 100, 200 or 280 yd rolls as required. In certain instances, particularly those using oil coolants, special grades of filter material, including fabric, may be employed. Operating costs under average conditions will not exceed a few shillings per week. Overall dimen-

sions of the four units are 53 in × 26 in, 60 in × 34 in, 66 in × 41 in, and 98 in × 62 in. Approximate height is 20 in. Larger units having flow rates of 4,000 gal/hr and over are built specially to meet specific requirements.

The equipment, it is claimed, makes possible a number of production economies. The complete removal of solids from the coolant virtually eliminates the cause of wheel clogging and thus less wheel dressing is required and the wheel life is considerably extended. Service reports show that life may be increased by 100 per cent, and rejects and spoilages be reduced. Since the wheel maintains its cutting surface longer, a quicker rate of stock removal is possible in some instances.

The coolant tank needs cleaning only four times a year, generally because the coolant becomes unhygienic, instead of the usual weekly cleaning of the normal settling tank. This not only reduces maintenance charges, but effects substantial savings in the coolant costs. Machine output should be improved as there is less down time for the purpose of wheel dressing and tank cleaning. The deposit on the filter medium leaves the clarifier in a semi-dry state and dries out quickly in the sludge bin. Thus valuable material can be salvaged easily for reclamation.

The clarifier can be usefully employed in connection with other machining operations. It is, of course, equally suitable for honing machines and for deep-hole boring machines. A 4,000 gal/hr unit has been supplied for use with a Carlstedt deep-hole borer. It can also be used in strip rolling to ensure the removal from the lubricant of solids likely to cause surface imperfections on the finished work.

CORRESPONDENCE

FRONT SUSPENSION

SIR,—Mr. D. R. Hume, in his interesting review of automobile front suspension systems, states that with the vertical pillar suspension layout, unless a splined pillar arrangement is used, the front wheels toe-out if there is any deflection from the static position of one wheel. I have in mind an alternative arrangement that ensures correct steering geometry under all conditions of wheel movement. This is the subject of a patent application so, of course, I cannot say more of it just now.

With regard to the trailing arm type of suspension, Mr. Hume does not mention one of the disadvantages. This is the alteration in wheelbase consequent upon the trailing arms moving about their pivots. The rise and fall of the wheel on a relatively small radius, leads to fluctuations in its rotational speed. Its inertia resists this acceleration and deceleration and thus causes the wheel to rotate the stub axle backwards and forwards about the king pin. This, in turn, naturally has some effect upon both adhesion and steering-wheel kick.

In connection with the divided axle suspension, Mr. Hume states that there is less track variation and wheel tilt than with any other system, provided the roll-centre height is 70 per cent or more of the tyre rolling-radius. In view of his previous remarks under the headings of Vertical pillar, Trailing arm and Dubonnet systems, this is difficult to understand. Perhaps he will explain.

He also mentions that with the divided axle system, the rate of tyre wear is noticeably low. This, it is agreed, is borne out by experience. I suggest, however, that one of the major reasons may be that the wheel does, in fact, follow the path that the designer intended when encountering road inequalities. The high rate of wear experienced with modern suspension systems is referred to in a recent paper by French and Gough at the Fifth International Technical Congress of the Motor Industry in Munich. The average front-tyre life for cars with and without independent front suspension is in the ratio of 75 to 110. Whether or not a part of this heavy rate of wear is due to deflections in structure, in particular in wishbones, wishbone bearings, wheels, steering connectings, etc., is questionable. It is thought that the front wheels fail, in many cases by quite large margins, to follow the path indicated by the layout on the drawing board. Examples to come to mind of cars which handle unusually well and with which a low rate of tyre wear is experienced: in these, the designer had evidently gone to some pains to ensure rigidity in the structure.

Finally, I think Mr. Hume is wrong in ascribing a tendency to give insufficient warning of impending breakaway to a coincidence of centre-of-gravity and roll-centre. Whilst, in some pre-war sports cars, this tendency was not unusual, there were others in which these centres must have been at least as close, but which, as a result of inherently good steering and handling qualities, would give the driver ample warning, with minimum of roll.

R. C. SYMONDSON A.F.C., M.A.,
West Horsley, Surrey.

Mr. Hume replies:—I shall be most interested to see the details, when they are published, of Mr. Symondson's device that enables the vertical pillar suspension arrangement to be used without there being track variation due to deflection of one wheel from the static position. Presumably it is simpler and less expensive than the splined pillar arrangement, but the roll-centre, no doubt, is still approximately at ground level.

Regarding the trailing arm suspension, it is, of course, true that with an arm of relatively small radius, slight fluctuations in rotational speed do occur as the wheel rises and falls, the magnitude being approximately proportional to the displacement. Under severe conditions there is a tendency for the reaction due to this acceleration to cause the stub-axle to oscillate about the king pin, and on at least one well known vehicle in which this form of suspension is employed a fibre washer damper is incorporated in the king-pin assembly to overcome that difficulty.

Regarding my statement that there is less track variation and wheel tilt with the divided axle system than any other, provided that the roll-centre height is at least 70 per cent of the tyre rolling-radius above ground level, the three systems Mr. Symondson mentions do not normally have a roll-centre height of more than approximately 10 per cent of the tyre rolling-radius and therefore are not relevant to this particular discussion.

If a double-transverse-link arrangement is designed so that it has a roll-centre at least 70 per cent of the tyre rolling- radius above the ground level, the wheel tilt and track variation, for the same general proportions, will exceed those for a divided-axle layout. The Ford vertical-leg arrangement will not, in the same circumstances, have greater wheel tilt but will give a greater track variation. These remarks are based on the assumption that the movements to the bump and rebound limits are normal, as also are the proportions of the suspension layouts.

I do feel that Mr. Symondson has raised some important points with regard to the lack of stiffness in the various suspension parts and its effects on tyre wear. But I also consider that this effect is only part of the trouble: much of the remainder is due to negative angles of attack of the tyres to the road during cornering. Some manufacturers, in an effort to reduce flexibility in suspension components, have eliminated rubber bearings. However, the proportions of the links in many of these arrangements are such that it is difficult to understand how they can but deflect under loads due to cornering.

Disregarding considerations relating to assembly, which should in any case take second place to the attainment of satisfactory performance, the attachment points of the links should be spread as much as is reasonably possible. This, of course, reduces the loads on the bearings and attachment points, as well as increasing the stiffness, with only a very small weight penalty. Two very good examples of this are on the new M. G. Magnette, and the Ford models with the vertical-leg type suspension. Rubber bearings are used throughout for actual suspension on both arrangements, except for one ball joint in the Ford scheme; this ball joint also serves as a steering pivot.

With regard to Mr. Symondson's final point. I feel that because of the general proportions of most of these sports cars, that is, very low centre of gravity, roll axis approximately parallel to the ground and some 80 per cent of the tyre rolling radius above the ground level, as in cars with beam axles, the optimum conditions for correct dynamic wheel, and consequently tyre, loadings are bound to be critical. The following example supports this contention. A certain car fitted with a standard body had a horizontal roll axis 10 in above ground level, and its centre of gravity height was 12 in. When a special body was fitted the centre of gravity was raised to a point 14 in above ground level. This increased the moment between the centre of gravity and the roll axis by 100 per cent. As was so often the case, the vehicle with the special body and the raised centre of gravity handled far better than the standard one.

Your correspondent also maintains that the pre-war cars handled well because of good inherent steering and handling qualities. One is prompted to ask, what is it that provides inherently good steering and handling qualities?

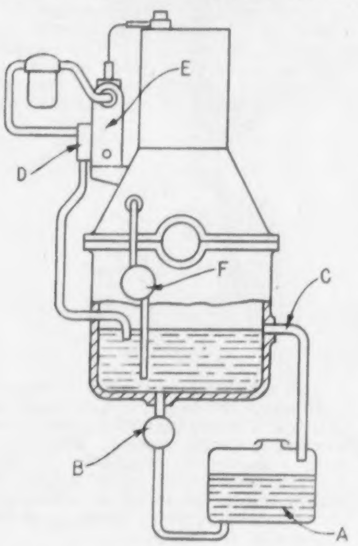
D. R. HUME,
East Twickenham, Middlesex.

CURRENT PATENTS

A Review of Recent Automobile Specifications

Common fuel and lubrication system

THE current development of lubricating oils of low viscosity has reached a stage at which such oils are essentially similar in character to fuel oils and can be used for either purpose in compression ignition engines. In a combined system, oil for fuel is drawn from the crankcase sump and the lubricant is continuously renewed by fresh oil. Thus it does not



No. 720222

become fouled by recirculation over a long period and the loss involved in draining and replenishing is wholly or partially avoided.

From a tank A at low level, oil is fed directly to the engine sump by a pump B and an overflow pipe C returns surplus to the tank. Oil for fuel is drawn off by feed pump D and delivered through a filter to the injection pump E. Lubricating oil is delivered to the engine by pump F in the usual manner. Presumably a filter is included in this supply line, although it is not shown in the diagram. The respective levels of the inlet pipes of the fuel and lubricating systems are arranged so that adequate lubrication is assured under all circumstances.

In two modified arrangements the oil level in the engine sump is maintained by a float-controlled valve. With a high-level tank the feed is by gravity, while for a low-level tank a pump with a spring-loaded relief valve and a by-pass circuit is used. *Patent No.* 720222. *C.A.V. Ltd.*

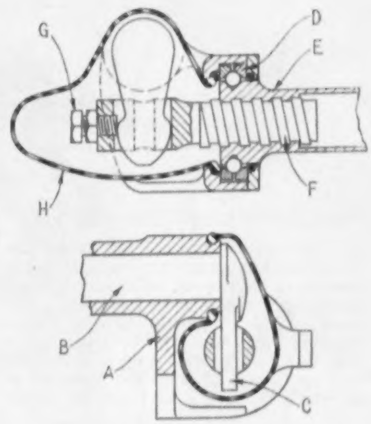
Cooling system

SINCE an engine cooling system must necessarily be capable of dissipating the maximum heat output under conditions of sustained full-throttle operation in tropical temperatures, it follows that when operated at part-throttle for any appreciable period the engine is over-cooled by reason of the surfeit of cooling capacity. It is the aim to correlate the heat-dissipating capacity of the system with the prevailing degree of throttle opening, so that relatively high operating temperatures obtain on part-throttle settings with the consequent benefit of a reduced rate of fuel consumption.

The cooling system proposed has a sectional radiator comprising a small capacity unit A, and a unit B of larger capacity which jointly are capable of dissipating the maximum heat output. Each unit has a separate header tank and inlet, but the two have a common bottom tank and outlet leading to the usual impeller C. This discharges to a gallery D and coolant is delivered to the zones of the respective exhaust valve seatings. The outflow pipe E leads to the header tank of radiator unit A and is provided with the usual thermostatic valve F. Upstream of the thermostat is a branch pipe to the header tank of radiator unit B and in this pipe is fitted a butterfly-type control valve. This valve is constrained to either the closed or open position by a lever arm and a toggle spring G and is influenced to either position by a lost-motion linkage connected with the accelerator pedal.

On starting from cold, coolant flow through unit A only is limited to the usual thermostat bleed and the coolant temperature rises rapidly until the thermostat opens. Up to about half throttle, circulation is through unit A only, but beyond that point movement of the accelerator pedal linkage causes the branch control valve to open gradually and the coolant flow to be increased. Still further depression of the accelerator pedal results in the control valve lever passing over centre and the spring G snaps the valve fully open. *Patent No.* 720330. *Morris Motors Ltd.*

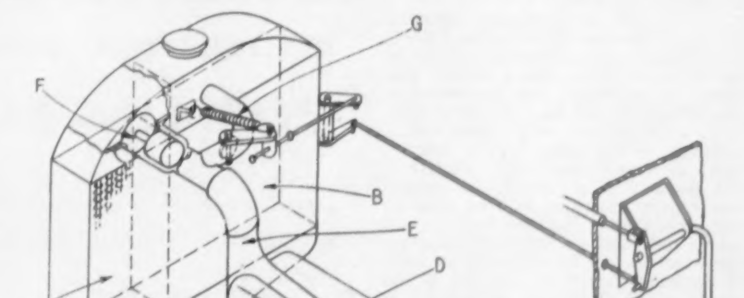
Screw-and-nut steering gear



No. 720207

THE aim of this invention is to provide a steering mechanism that is of light weight and is simple to produce and assemble. A main bracket A, adapted to be secured to the chassis framing, supports the spindle B having at one end a radial arm C. On the other end of spindle B is mounted the drop arm or lever connecting with the steering linkage of the road wheels. In a housing formed in the main bracket are mounted the coned races D of an anti-friction ball bearing supporting a rotatable nut E on an axis arranged at right angles to that of spindle B in a parallel plane. The nut is formed with or secured to the steering wheel shaft.

Engaged in the nut is an axially movable screw F, the free end of which is slotted to receive the end of arm C. This end is shaped to an involute form so that in all radial positions of the arm its two points of contact with the ends of the slot will lie in or close to a plane containing the axis of the screw. For initial adjustment, and to take up slackness resulting from wear, a screwed abutment G is fitted in the end of screw F. Enclosure of the mechanism is by means of a moulded rubber cover H. *Patent No.* 720207. *Burman and Sons Ltd. and W. H. Briggs.*



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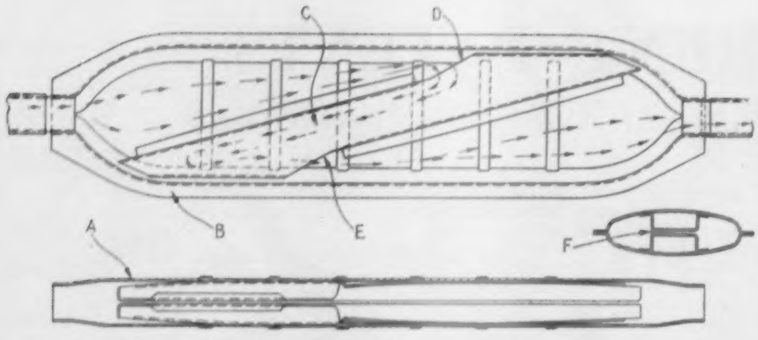
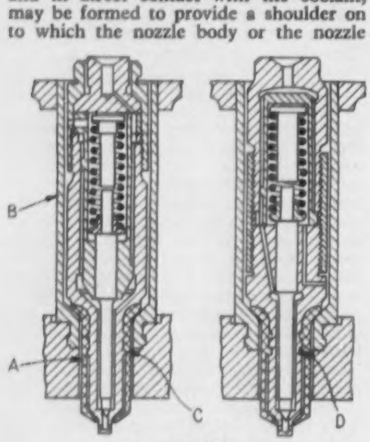


No. 719975

Heat-shield for fuel injectors

IN attempts to improve the conduction of heat away from the nozzle of a fuel injector, a variety of constructional and mounting arrangements have been devised. The copper or bronze sleeve, expanded into the cylinder head bosses and in direct contact with the coolant, may be formed to provide a shoulder on to which the nozzle body or the nozzle



No. 719952

cap nut is seated. In the case of "long reach" injectors, the long stem portion of the nozzle body may be shielded from heat conducted or radiated from the combustion chamber by a tubular extension of the cap nut. Since considerations of strength preclude the use of metals other than steel for the nozzle body and the cap nut, these expedients may not be completely effective.

According to the invention, optimum conditions are obtained by rolling a sleeve A of a good heat-conducting material on to the stem of the nozzle body. This makes direct contact with the seating of the injector in the end of the cylinder head sleeve B, and at its inner end is spaced from the nozzle body in order to serve as a radiation shield. Preferably the sleeve A is of copper, the thermal conductivity of which is from eight to ten times that of the steel used for the nozzle body, and can thus be of relatively small section. At the extremity, sleeve A is formed to enshroud the end of the nozzle as far as possible so that only a minimum area of the nozzle is directly exposed to the heat of the combustion chamber.

Two forms of construction are shown, with the sleeve secured to the nozzle stem by rolling over ribs C in one instance and into grooves D in the other. *Patent No.* 719952. *S.A. Adolphe Saurer (Switzerland).*

Silencer construction

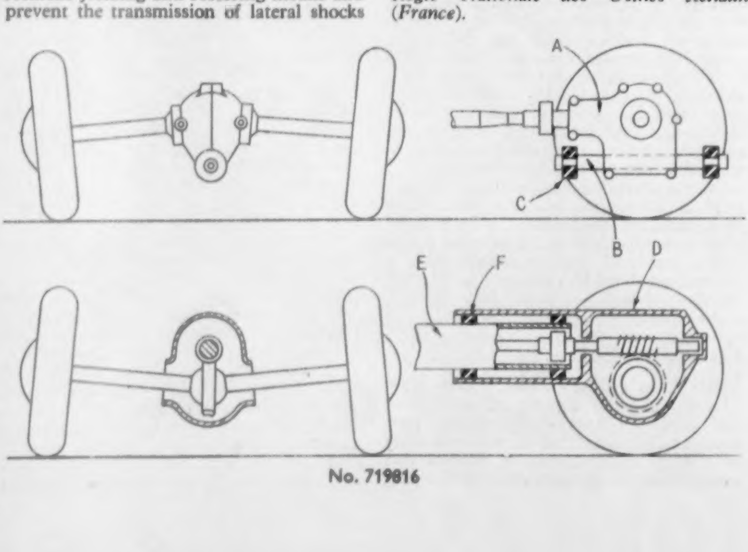
A SHALLOW silencer of elliptical cross section is formed of two pairs of sheet metal pressings united by spot welding. The casing comprises two similar pressings A, having peripheral flanges B. Into each of these pressings is spot welded a flanged, channel section baffle C located diagonally. Each of these baffles divides its associated half-casing into three communicating ducts, the connection being by way of apertures D and E in the walls of the baffle. When the casing halves are assembled by spot welding the flanges B, a narrow space F remains between the two baffles to permit expansion.

The exhaust gases pass through the silencer in three streams. One stream from the inlet flows along the casing to aperture D, reverses direction to pass through the baffle, reverses direction again at aperture E and flows along the casing to the outlet. A second stream makes a similar passage in the other half-casing and the third stream, only a relatively small proportion of the total flow, passes directly from inlet to outlet by way of the space F between the baffles. *Patent No.* 719975. *Vauxhall Motors Ltd.*

Resiliently mounted differential casing

IN a vehicle having a chassis-mounted differential gear and swinging half-axles, the differential casing is arranged to oscillate about a substantially horizontal longitudinal axis outside the horizontal plane containing the points of pivotal mounting of the half axles. Rubber springs at the oscillation point provide a resilient yielding and restoring means and prevent the transmission of lateral shocks from the wheels of the vehicle body.

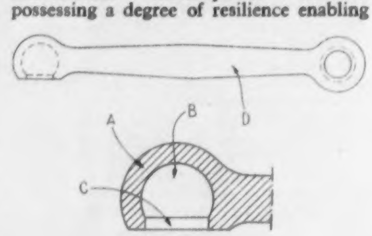
Two constructional arrangements are shown. In one, differential casing A is rigidly mounted on a shaft B, the protruding ends of which are supported in annular torsional rubber springs C attached to the vehicle frame. In the other, differential casing D has a tubular extension by which it is mounted over the end of a tubular frame member E with interposed rubber springs F. Pivotal mounting of the half axles below the oscillation axis of the differential casing, as in the second example, is of special advantage in the case of half axles having normally a negative inclination. *Patent No.* 719816. *Daimler-Benz A.G. (Germany).*



No. 719816

Separable ball-and-socket joints

IN a known type of joint a separate resilient ring constricting the mouth of the socket retains the ball in position and is deformable to permit rapid assembly or dismantling of the joint. The present invention proposes a socket made of a conventional thermoplastic material possessing a degree of resilience enabling



No. 718968

it to be snapped over the ball to engage or disengage the joint. It also relates to a method of producing such a socket by an injection moulding process. Instead of using a multi-part, detachable core for the re-entrant cavity, a solid core is employed and the socket is withdrawn after moulding by taking advantage of the elasticity of the material.

The socket A may be produced as a unit for attachment by any means to a rod or lever. Cavity B, which may be cylindrical, cylindro-conical, bi-conical or spherical in shape, has a diameter slightly greater than that of the ball-end while the throat C is slightly smaller than the ball diameter. Alternatively, one or more sockets may be formed integrally with a rod or link of thermoplastic material, as at D. *Patent No.* 718968. *Regie Nationale des Usines Renault (France).*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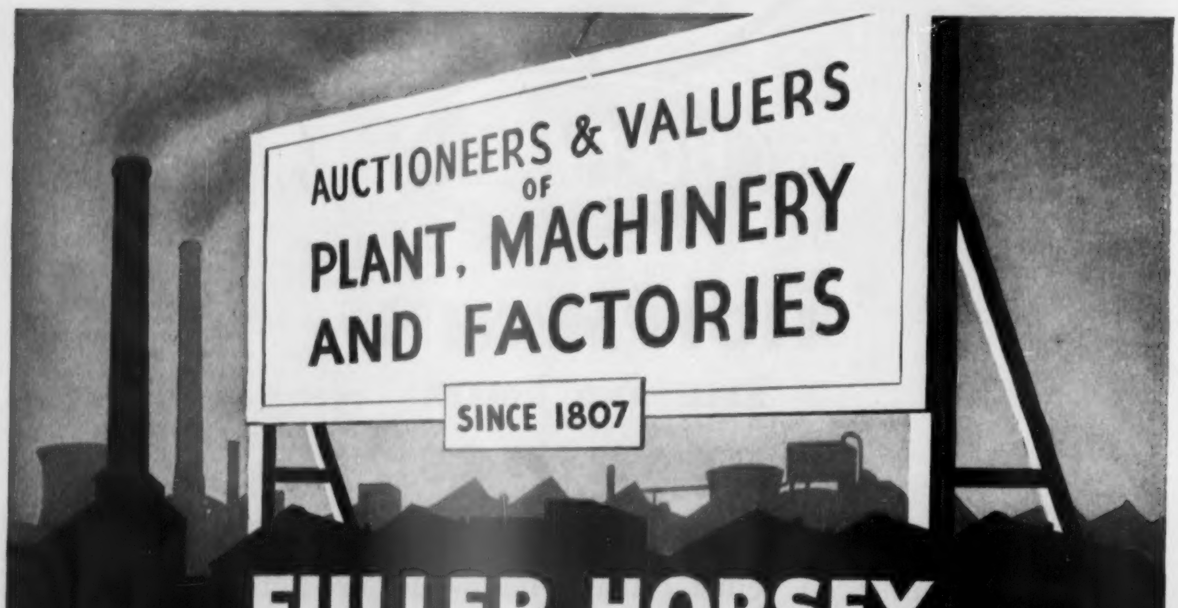
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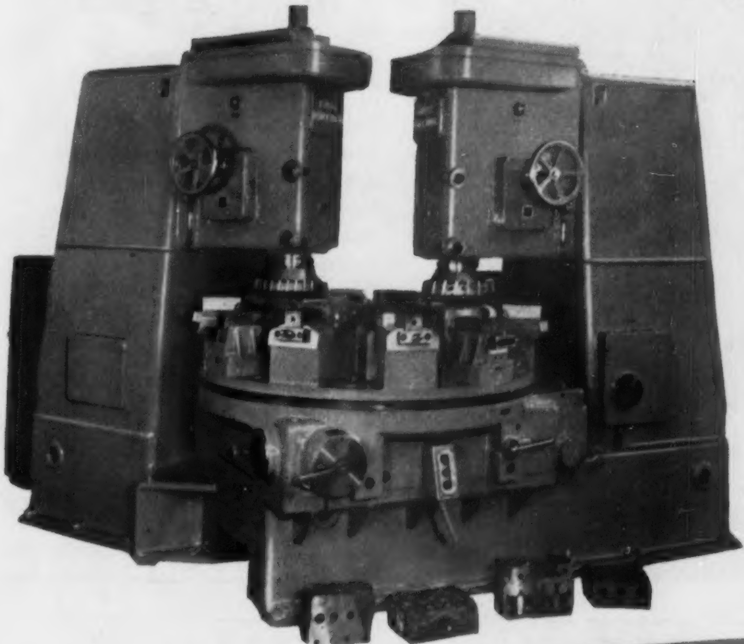

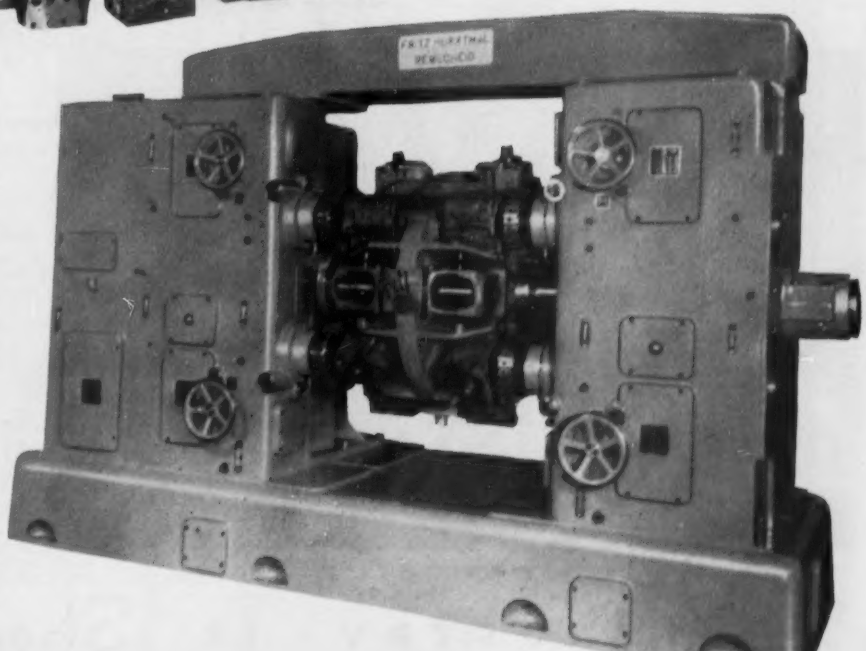

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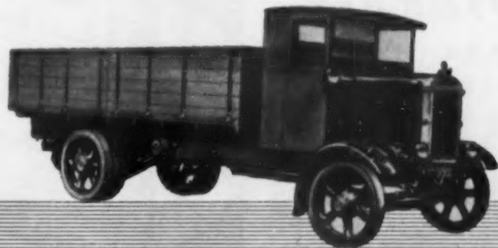

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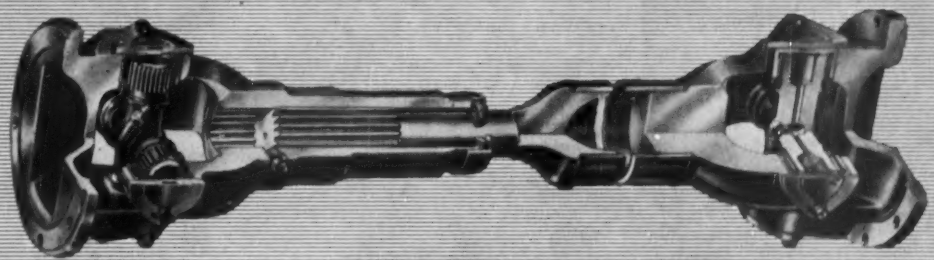

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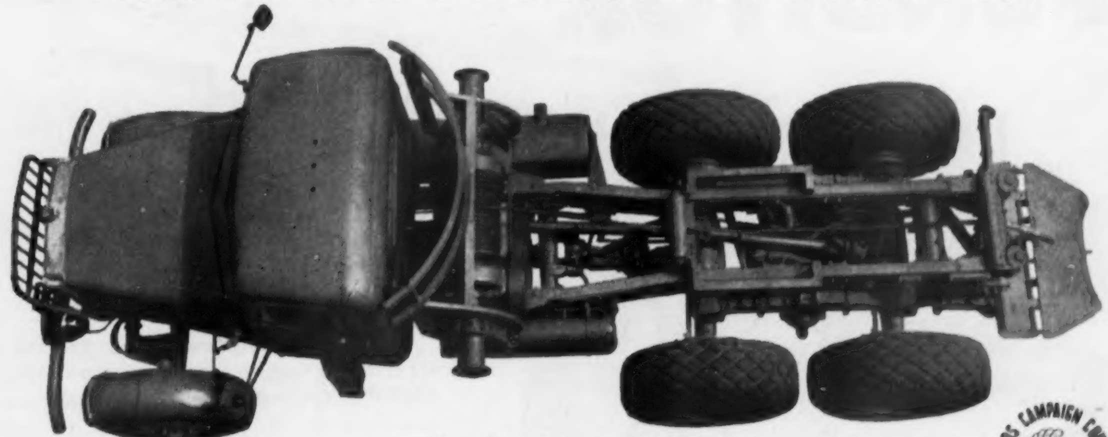

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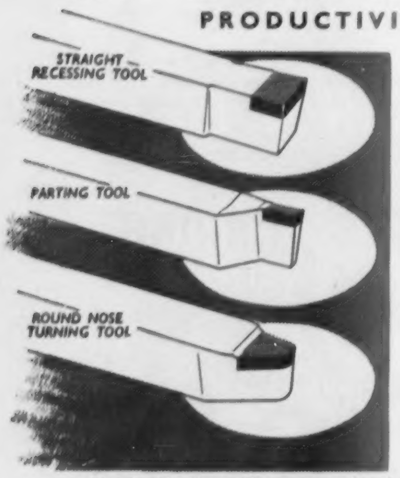

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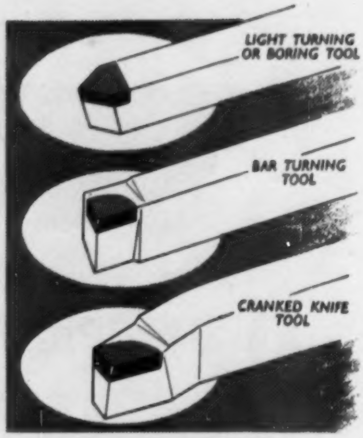

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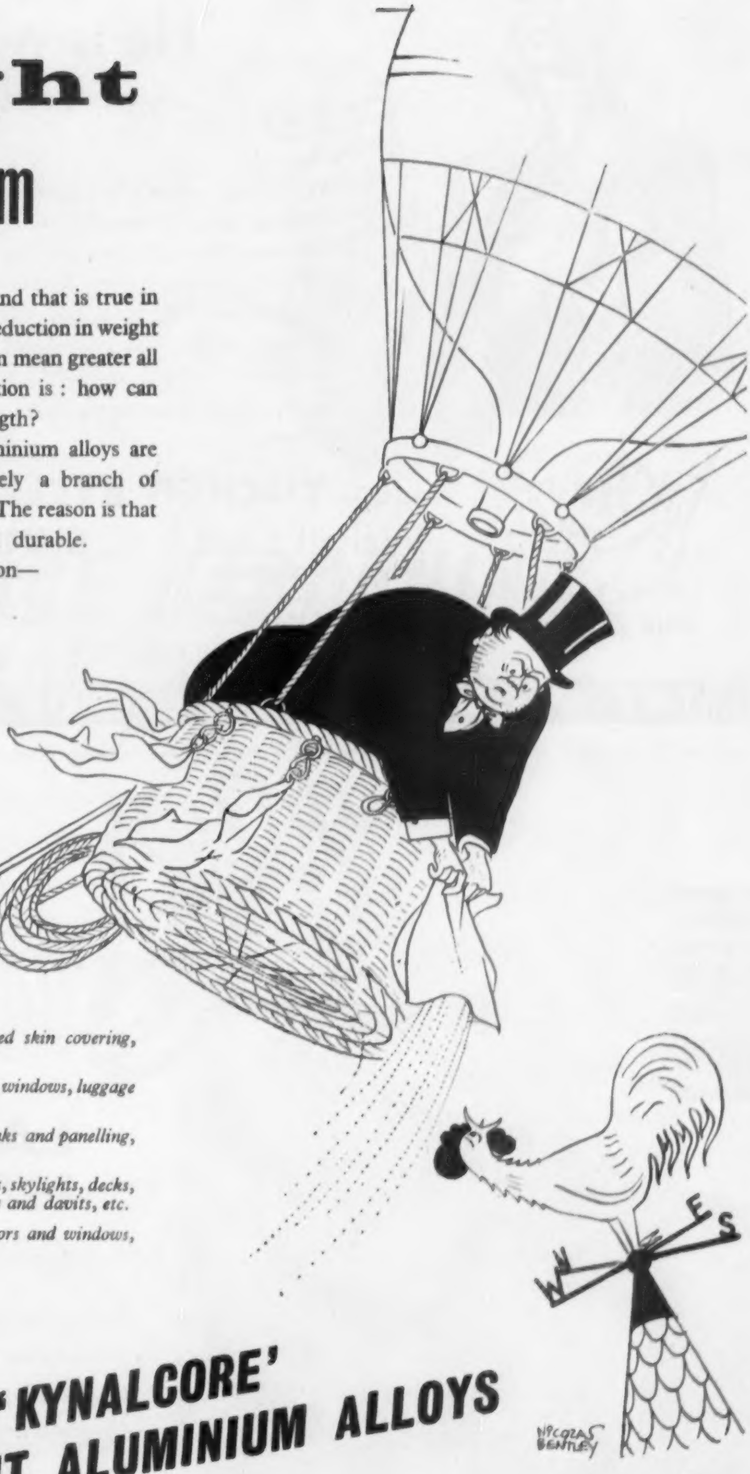

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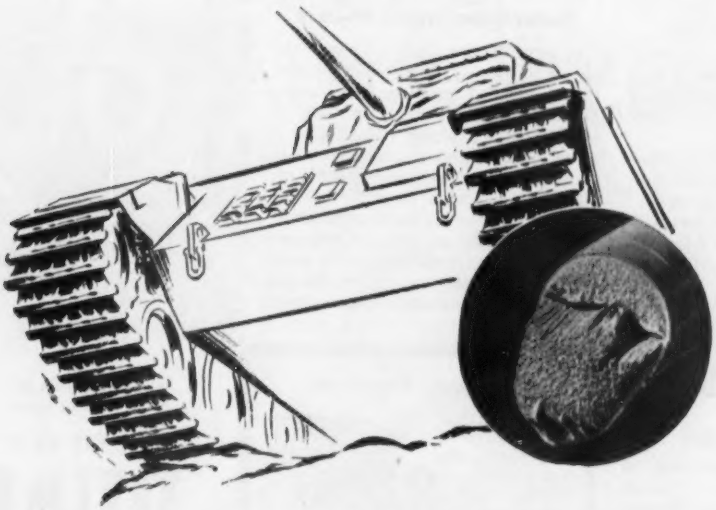

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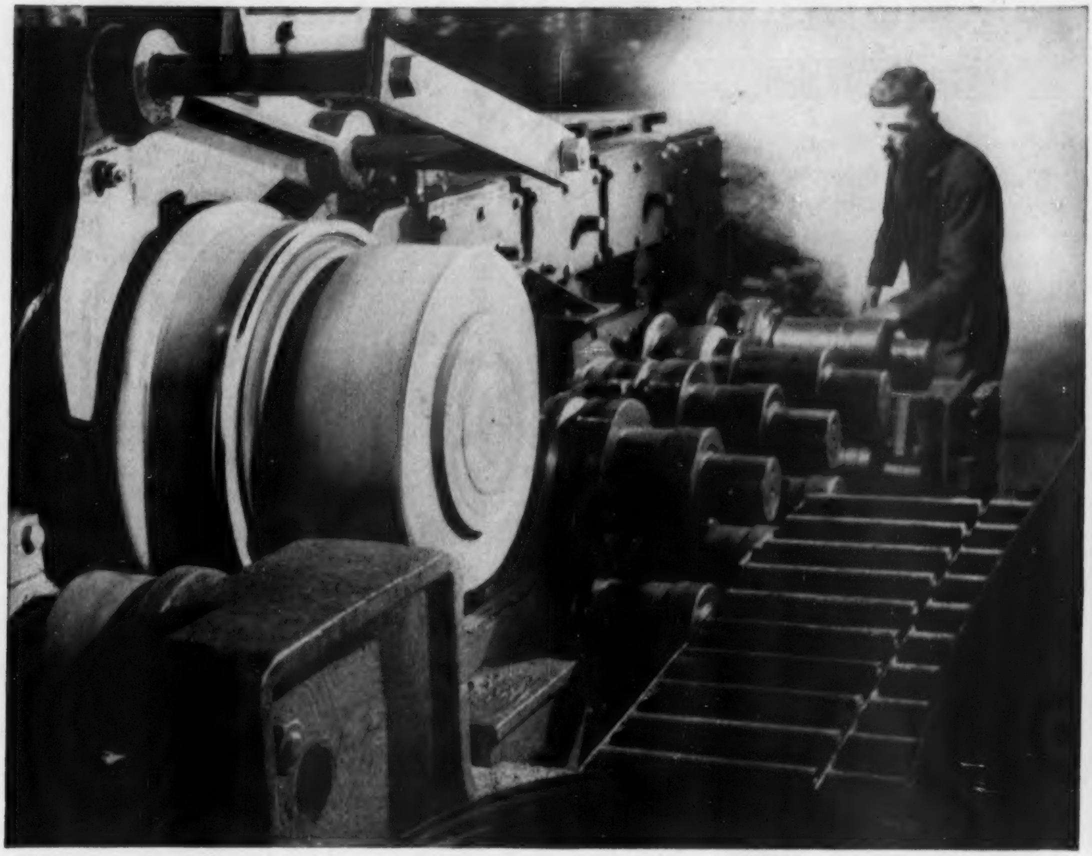

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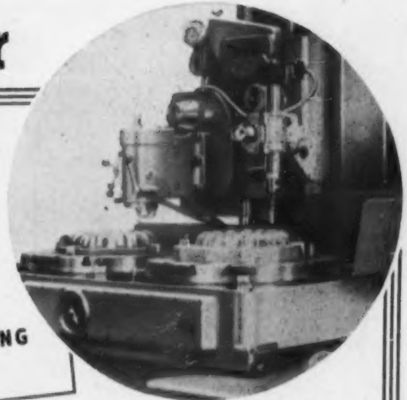

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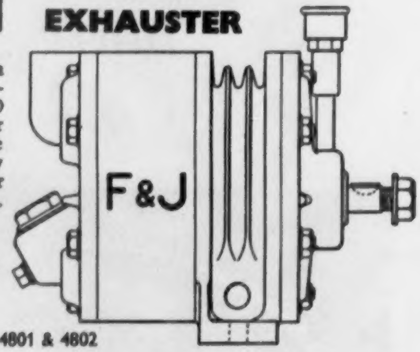

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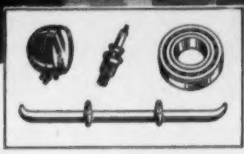

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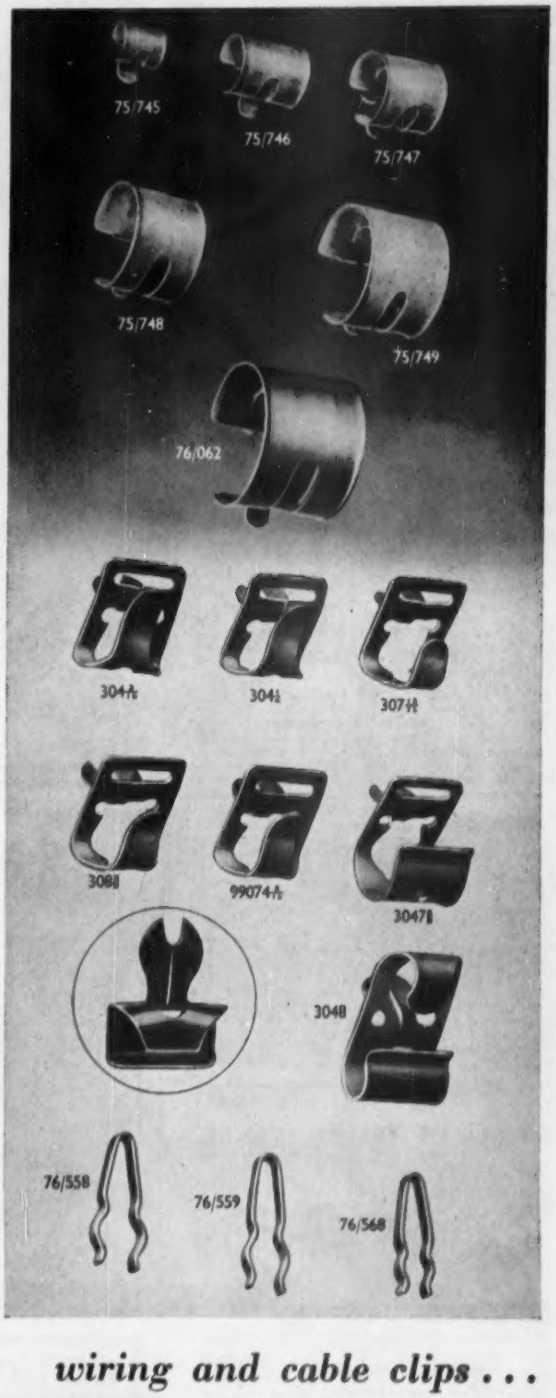

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OBITUARY

JOHN HOLROYD & Co. Ltd., announce with great regret the sudden death of their London Area Manager, Mr. Roland H. Hudson, on Sunday the 17th July 1955, whilst on holiday in Majorca [5055

PATENTS

THE proprietor of British Patent No. 577425, entitled "Improvements in Spring Suspension for Vehicles," offers same for licence or otherwise to ensure its practical working in Great Britain. Enquiries to Singer, Stern & Carlberg, Chrysler Building, New York 17, N.Y., U.S.A. [5035

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